THE INSTITUTION OF MECHANICAL ENGINEERS

General Discussion on Lubrication & Lubricants

THE INSTITUTION OF MECHANICAL ENGINEERS

PROCEEDINGS OF THE GENERAL DISCUSSION ON LUBRICATION & LUBRICANTS

13th–15th October 1937

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of Volume 1

GENERAL DISCUSSION ON LUBRICATION AND LUBRICANTS

Introduction

The General Discussion on Lubrication and Lubricants held at the Central Hall, Westminster, from the 13th to 15th October 1937, was organized by The Institution of Mechanical Engineers with the object of reviewing the present state of the science and the practice of lubrication in order to correlate theory and pure research with practice, to consider methods of bearing design, to obtain current views upon bearing metals, and to bring out the significance of laboratory tests, including wear and friction tests.

The Discussion was organized by the Institution with the co-operation of the following scientific and technical bodies:—

British :—

Belfast Association of Engineers.
British Association of Refrigeration.
British Engineers' Association.
Faraday Society.
Institute of Fuel.
Institute of Marine Engineers.
Institute of Metals.
Institute of Physics.
Institute of Transport.
Institution of Automobile Engineers.
Institution of Chemical Engineers.
Institution of Civil Engineers.
Institution of Civil Engineers of Ireland.
Institution of Electrical Engineers.
Institution of Engineers and Shipbuilders in Scotland.
Institution of Heating and Ventilating Engineers.
Institution of Locomotive Engineers.
Institution of Mining Engineers.
Institution of Naval Architects.
Institution of Petroleum Technologists.
Institution of Production Engineers.
Iron and Steel Institute.
Junior Institution of Engineers.
Manchester Association of Engineers.
North East Coast Institution of Engineers and Shipbuilders.
Physical Society.
Royal Aeronautical Society.
Society of Chemical Industry.
Society of Engineers.
South Wales Institute of Engineers.
Textile Institute.

Dominion and Foreign :—

Australia (Institution of Engineers).
Austria (Österreichischer Ingenieur- und Architecten-Verein).
Belgium (Belgian Standards Association).
Canada (National Research Council).
Czechoslovakia (Spolek Ceskoslovenskych Inzenyru).
France (Société des Ingénieurs Civils de France).
France (Société des Ingénieurs de l'Automobile).
Germany (Verein Deutscher Ingenieure).
Germany (Wirtschaftsgruppe Kraftstoffindustrie).
Holland (Koninklijk Instituut van Ingenieurs).
India (The Institution of Engineers).
Italy (Sindacato Nazionale Fascista Ingegneri).
Japan (Society of Mechanical Engineers).
New Zealand (New Zealand Society of Civil Engineers).
Norway (Norske Ingeniorforeningen).
South Africa (South African Institution of Engineers).
Sweden (Svenska Teknologforeningen).
Switzerland (Société Suisse des Ingénieurs et des Architectes).
United States (American Petroleum Institute).
United States (American Society for Testing Materials).
United States (American Society of Mechanical Engineers).
United States (Society of Automotive Engineers).

The Committee appointed by the Institution to organize the Discussion was constituted as follows :—

Dr. H. J. Gough, M.B.E., F.R.S., M.I.Mech.E. (*Chairman*).
Mr. E. A. Evans.
Sir H. Nigel Gresley, C.B.E., D.Sc., M.I.Mech.E.
Mr. H. L. Guy, F.R.S., M.I.Mech.E.
Mr. Aubrey B. Smith, M.I.Mech.E.
Professor H. W. Swift, M.A., D.Sc. (Eng.), M.I.Mech.E.

To render practicable the adequate discussion of the papers within the limited time available, summaries of the papers in each Group were prepared by the following :—

For Group I. "Journal and Thrust Bearings": Professor H. W. Swift, M.A., D.Sc. (Eng.), M.I.Mech.E.

For Group II. "Engine Lubrication":—

Internal Combustion Engines: Mr. Harry Ricardo, B.A., F.R.S., M.I.Mech.E.

Reciprocating Steam Engines: Mr. W. A. Stanier, M.I.Mech.E.

For Group III. "Industrial Applications": Lt.-Colonel S. J. M. Auld, O.B.E., M.C., and Mr. E. A. Evans.

For Group IV. "Properties and Testing": Dr. H. J. Gough, M.B.E., F.R.S., M.I.Mech.E.

Altogether 136 papers were presented.

Over 600 members and visitors registered for the meetings and the average attendance at the four sessions of the Discussion was about 500 members and visitors.

To illustrate the subjects under discussion, it was decided to arrange an Exhibition devoted to lubricants, lubricators, bearings, bearing materials, and testing and research apparatus. The Committee which organized the Exhibition was composed as follows:—

Mr. E. A. Evans (*Chairman*).
Mr. O. F. Brown.
Dr. F. H. Garner.
Mr. J. Kewley.
Lt.-Colonel K. G. Maxwell.
Mr. David E. Roberts, M.I.Mech.E.
Mr. G. F. Westcott, A.M.I.Mech.E.

The Exhibition was held at the Science Museum through the courtesy of the Director, Colonel E. E. B. Mackintosh, D.S.O.

A Conversazione was held at the Science Museum on 13th October, the members and visitors being received by Major General A. E. Davidson, D.S.O., M.I.Mech.E. (*Past-President*), who also opened the Exhibition. In the course of his opening speech, Major General Davidson said that they would hear with great regret that the indisposition of the President, Sir John E. Thornycroft, K.B.E., had prevented him from opening the Discussion on Lubrication and also from welcoming them at the Conversazione. He felt sure that those present would like him to convey to the President the expression of their regret at his enforced absence. Continuing, Major General Davidson said that that afternoon, Sir Nigel Gresley, in the absence of the President, opened a most important series of discussions and thereby a new leaf in a very old ledger, namely "The History of Studies on Lubrication" by The Institution of Mechanical Engineers. As those studies started as early as 1883 with Beauchamp Tower's first report, they had long passed the Jubilee of such work. It was therefore a suitable occasion to take stock of the many years of labour which the Institution had undertaken on a subject of such interest and vital importance to all engineers. The Institution had never lost sight of the value of scientific research. Physicists and scientists must progress hand-in-hand with engineers so that they could help to solve engineering problems as they arose. No engineer could design a successful piece of mechanism without giving due consideration to its lubrication, and no user could expect good service from his machinery without paying due attention to that subject.

It was clear that the subject of lubrication was of importance to a very large number of non-engineers who relied upon various types of machines to help them in their every-day life, whether they were users of motor cars, sewing machines, lawn mowers, or typewriters.

In his experience, one of the greatest advances in lubrication had been the application of automatic systems to various types of machinery and plant. It had long been realized that most machines were sold to non-technical users who had little time, desire or skill to deal with the lubrication of their machines, and it had become essential that machines should operate without any attention for long periods by means of a simple process. Possibly, the earliest practical appreciation of that principle was the Bellis forced-lubrication engine which was produced by a member of the Institution whose descendant, Mr. John Bellis, was a member of the present Council.

The Council had considered it advisable to implement the General Discussion on Lubrication by arranging an Exhibition illustrating many of the features dealt with in the papers. He had no intention of enumerating the many exhibits, but the very carefully chosen collection showed in a variety of ways how lubrication had developed. Some of the older exhibits now seemed rudimentary and he felt sure that the ladies who were present would be interested to see the development of the early sewing machines compared with those made to-day. The Exhibition would remain open for another three weeks, and the general public as well as the present gathering would find it of great interest.

Those who only came to see some specific exhibit should be reminded that they would also find much of interest in the remainder of the Museum. That reminded him that the last occasion on which he had the pleasure of welcoming the members in the Science Museum was the James Watt bi-centenary ceremonial.

In conclusion, General Davidson said that if the nations could be persuaded to discuss affairs in the spirit in which engineers of all nations were prepared to deal with engineering developments, great advance towards the solution of world problems, many of which were, in essence, due to new developments, would be achieved.

General Davidson expressed his appreciation of the presence of numerous ladies and visitors and, after he had conveyed the thanks of the Institution to Colonel E. E. B. Mackintosh, D.S.O., the Director of the Science Museum, for having permitted the Exhibition to be held there, and to the Exhibition Committee and the exhibitors for having provided such a comprehensive range of exhibits, he declared the Exhibition open.

The Exhibition remained open to the public until 31st October, the total attendance being 18,300.

Thanks are due to the following for having provided or arranged for exhibits:—

The Air Ministry.
Institution of Petroleum Technologists.
Institution of Automobile Engineers.
Dr. F. P. Bowden (University of Cambridge).
Professor H. W. Swift (University of Sheffield).
Dr. H. J. Gough, F.R.S. (National Physical Laboratory).
Professor G. I. Finch (Imperial College of Science and Technology).
The Bradford Technical College.
Messrs. A. A. Clark and H. J. Hodsman (University of Leeds).
Mr. S. A. de Lacy.
Professor R. V. Southwell, F.R.S. (University of Oxford).
Mr. J. H. Wells.
E. G. Acheson, Ltd.
Alfa-Laval Company, Ltd.
W. H. Allen, Sons and Company, Ltd.
Anglo-American Oil Company, Ltd.
Anglo-Iranian Oil Company, Ltd.
Austin Motor Company, Ltd.
Autoklean Strainers, Ltd.
Automotive Products, Ltd.
W. and T. Avery, Ltd.
Sir W. H. Bailey and Company, Ltd.
Bakelite, Ltd.
Bristol Aeroplane Company, Ltd.
British Northrop Loom Company, Ltd.
British Timken, Ltd.
David Brown and Sons (Hudd.), Ltd.
Bruntons (Musselburgh), Ltd.
Daimler Company, Ltd.
Davidson and Company, Ltd.
Alexander Duckham and Company, Ltd.
Dunlop Rubber Company, Ltd.
Ellison Insulations, Ltd.
English Electric Company, Ltd.
Glacier Metal Company, Ltd.
Great Western Railway Company.
Alfred Herbert, Ltd.
Hoffmann Manufacturing Company.
Hoyt Metal Company of Great Britain, Ltd.
Imperial Chemical Industries, Ltd.
International Technical Developments, Ltd.
London and North Eastern Railway Company.
Joseph Lucas, Ltd.
Luvax, Ltd.
Manganese Bronze and Brass Company, Ltd.
Metafiltration Company, Ltd.
Metropolitan-Cammell Carriage and Wagon Company, Ltd.
Metropolitan-Vickers Electrical Company, Ltd.
Michell Bearings, Ltd.
Mirrlees, Bickerton and Day, Ltd.
Morgan Crucible Company, Ltd.
D. Napier and Son, Ltd.
North Metropolitan Electric Power Supply Company.
C. A. Parsons and Company, Ltd.
F. Perkins, Ltd.
Petters, Ltd.

Peyinghaus Iron and Steel Works (Mr. V. A. Bary).
Pressed Steel Company, Ltd.
Ransome and Marles Bearing Company, Ltd.
Reavell and Company, Ltd.
Rolls Royce, Ltd.
Sheepbridge Stokes Centrifugal Castings Company, Ltd.
Shell-Mex and B.P., Ltd.
Simms Motor Units, Ltd.
Skefko Ball Bearing Company, Ltd.
S. S. Smith and Sons (Motor Accessories), Ltd.
Snowdon Sons and Company, Ltd.
Southern Railway Company.
Stream-Line Filters, Ltd.
Super-Centrifugal Engineers, Ltd.
Swan, Hunter, and Wigham Richardson, Ltd.
Clement Talbot, Ltd.
Tecalemit, Ltd.
Union Special Machine Company, Ltd.
Vokes Ltd.
C. C. Wakefield and Company, Ltd.
Walker Brothers (Wigan), Ltd.
Leonard Williams and Company, Ltd.

An interesting series of historical exhibits was loaned by the Science Museum, so that the Exhibition provided a valuable survey of the whole subject of lubrication and lubricants.

LIST OF PAPERS IN GROUP I

LIST OF PAPERS IN GROUP II

GROUP I. JOURNAL AND THRUST BEARINGS

THE PERFORMANCE OF OIL RINGS

By E. Baildon, M.Sc.(Eng.), A.M.I.Mech.E.*

The oil ring was developed for the lubrication of journal bearings some 70 years ago, but not until comparatively recent years have the principles which govern its operation received scientific attention. The problem of oil ring propulsion presents analytical difficulties not greatly differing from those of journal bearing lubrication, and even less tractable is the problem of oil transfer from ring to journal which depends mainly on the adhesion and cohesion of the oil, properties which are at present imperfectly understood.

Experimental investigations undertaken in various countries† have yet scarcely passed an introductory stage; nevertheless, the practical significance of the available results is clearly discernible. It is the object of this paper to outline briefly a descriptive explanation of oil ring operation and to summarize such conclusions drawn from experimental results as may afford guidance to the designer of ring-oiled journal bearings.

Propulsion. Oil ring propulsion depends on the action of the moving journal surface operating on the inner surface of the ring in the region of their nearest approach. This action is normally transmitted to the ring by a separating film of oil and is therefore viscous in character; though in circumstances when the form of the ring's inner surface leads to break-down of the intervening oil film, frictional propulsive conditions analogous to those of "boundary" lubrication may be established.

Viscous propulsion requires that the ring shall always "slip" on the journal and the propulsive force P is then proportional to the difference $(V-v)$ between the surface speeds of the journal and the inner periphery of the oil ring. A rational expression, dimensionally satisfactory, for this force may be written $P=C.\lambda(V-v)s/t$, where λ is the viscosity, s and t represent the area and thickness of the propulsive film, and C is a constant.

Resistance to oil ring rotation is due to: (1) viscous drag on the ring surfaces passing through the oil bath and the meniscus raised from the oil bath surface; (2) the unbalanced weight of oil adhering to the rising

* Assistant Lecturer in Mechanical Engineering, Bradford Technical College.
† See References, p. 7.

portion of the ring; and (3) contact (if any) of the ring with stationary surfaces of the bearing. This resistance increases with ring speed and, for any given speed of the journal, the ring runs at that speed which produces equilibrium between the propulsive and resistive actions.

The resistive force F is composed partly of viscous tractive components proportional to the ring speed v and partly of inertia forces which depend on the rate at which the ring imparts momentum to the oil set in motion. Hence the resistive force may reasonably be expected to satisfy an expression of the form $F=a(A\lambda v/p+B\rho v^2)$, where ρ is the density of the oil, p is the perimeter of the cross-section of the ring, a is the area of the immersed surface of the ring, and A and B are constants. By equating the expressions for P and F, the journal speed

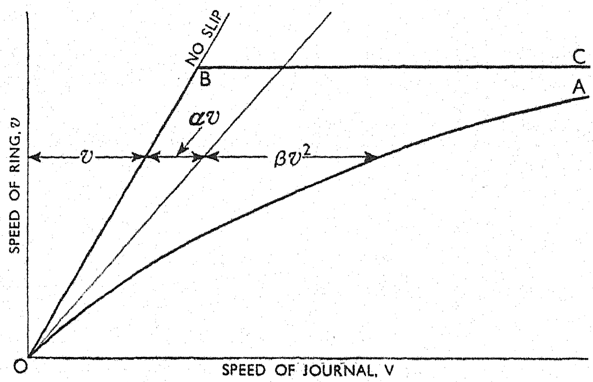

Fig. 1. Relationship between Journal and Ring Speeds

V may be found as a function of the ring speed v in the form $V=v(1+\alpha)+\beta v^2$, where $\alpha \propto at/ps$ and $\beta \propto \rho at/\lambda s$.

Some indication of the probable speed relationship under boundary propulsion may be afforded by assuming that the laws of solid friction are obeyed. Under such circumstances, at low journal speeds the frictional propulsive force adjusts itself to equilibrate the resistive force and no ring slip occurs; at some journal speed the frictional force becomes limiting; and at all higher journal speeds, the ring moves with constant maximum speed and ring slip of course takes place. Hence, the relationship between journal and ring speeds under viscous and solid frictional propulsive conditions may be expressed graphically by means of the curve OA and characteristic OBC respectively, shown in Fig. 1. Curves of actual oil ring speeds are shown in Fig. 2.

Oil Transfer. The transfer of oil from ring to journal, when not assisted by scraping or scooping devices, depends on the ability of

the journal to retain by adhesion oil with which it is brought into contact. Centrifugal forces due to journal and ring rotation hinder oil transfer by reducing or preventing gravitational flow to the journal from oil films on the sides of the ring and, at higher speeds, by overcoming adhesion. The film adhering to the inner surface of the ring forms the most important potential source of oil supply to the journal. As ring and journal approach, some oil from this film is exuded at the ring sides, but normally a thin film remains. The ring deposits part

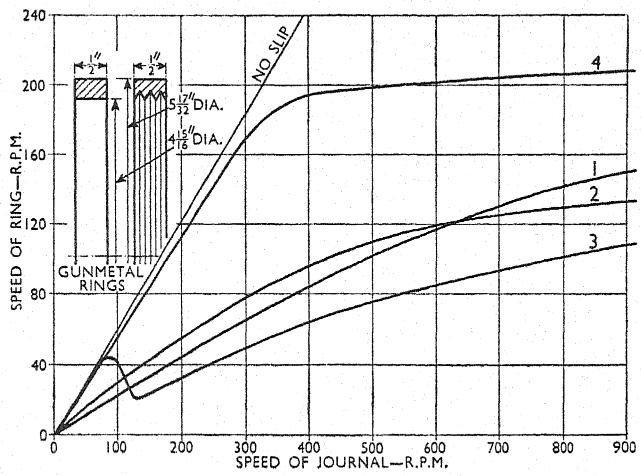

Fig. 2. Journal and Ring Speeds

Journal, 3 inches diameter
Curve 1 Plain ring, machine oil
,, 2 Grooved ring, ,, ,,
,, 3 Plain ring, light spindle oil
,, 4 Grooved ring, ,, ,,

of this film on the journal, which is able to retain a thin film of oil even at high speeds of rotation.

At low journal speeds, the exuded oil unites with that flowing under gravity from the sides of the ring to form thick streams which occupy the corner space between the sides of the ring and the surface of the journal. On recession of journal and ring, the streams divide, part adhering to the journal and the remainder to the ring. Beyond a certain critical range of journal speeds, centrifugal action reduces considerably the quantity of oil which can be retained by the journal from the side streams. This critical range of speeds is inversely proportional to the square root of the journal diameter. In recent experiments (Baildon 1937), the critical range was found to vary from about

230–260 r.p.m. for a 3-inch journal to about 110–125 r.p.m. for a 12-inch journal when an average machine oil was used. At still higher journal speeds, the transferred oil consists almost entirely of that acquired from the propulsive film.

Oil transfer thus takes place from: (1) the propulsive film at all journal speeds; and (2) the side-exuded oil and that which flows under gravity from the side films of the ring, more particularly at low journal speeds. While the thickness of the film acquired by the journal from the first source probably changes but little with journal speed, the thickness of the film acquired from the second source varies considerably.

The curve in Fig. 3 shows descriptively the relationship between the oil quantity transferred by adhesion per journal revolution and the journal speed. Actual oil quantities delivered by rings are shown

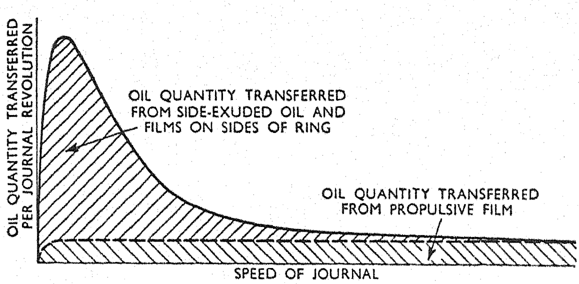

Fig. 3. Transfer of Oil by Adhesion in Relation to Journal Speed

graphically in Fig. 4. In order that an introductory conception of oil ring performance may be formed, these results are also expressed in terms of the thickness of the uniform film spread on the journal surface in a bearing in which the ratio of breadth to diameter=3/2. On such a criterion, of course, is oil ring performance most properly judged.

Ring Speed. The thickness of the films which adhere to the ring increases with ring speed and, provided that the journal speed is below the critical range, the smaller the proportion of ring slip the greater is the rate of oil transfer. At higher journal speeds, however, the advantages of thicker adhering films due to small slip are neutralized by greater centrifugal action and therefore, so far as adhesive transfer is concerned, the proportion of ring slip is of considerably less importance.

Proportions of Ring and Journal and Cross-Section of Ring. The thickness and length of the propulsive film is controlled to a large extent by the difference in the curvature of ring and journal. The

smaller the ratio of ring to journal diameter, the greater is the length and thickness of this film and the more favourable are the conditions for adhesive oil transfer to the journal at all journal speeds.

At low journal speeds when the side films of the ring contribute substantially to the amount of oil transferred, the radial thickness of the ring, which controls the side areas available for carrying oil into the transmission region, has probably more effect on performance than the axial width of the ring. At journal speeds higher than the critical range, however, what may be termed "side" oil transfer becomes negligible, the advantages of large radial thickness of ring are

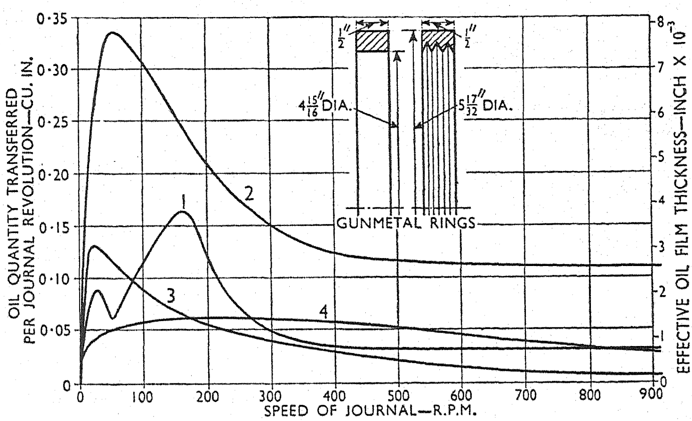

Fig. 4. Actual Deliveries of Oil by Rings

Journal, 3 inches diameter
Curve 1 Plain ring, machine oil
„ 2 Grooved ring, „ „
„ 3 Plain ring, light spindle oil
„ 4 Grooved ring, „ „
Rings immersed 1·3 inch in oil bath

lost and the transmission rate is roughly proportional to the width of the ring.

Material. Experiment (Baildon 1937) suggests that cast iron, steel, and gun-metal are satisfactory materials for the construction of oil rings. No particular advantages accompany the use of heavier materials such as lead, and lighter rings, of metals such as aluminium, are unreliable in operation.

Collecting Groove. The arc of recession of journal and ring is normally spanned by meniscal films from the sides of the ring to the journal surface. The circumferential extent of these films varies with

speed, but at relatively low journal speeds they extend to the collecting groove situated on the horizontal diameter. Oil then flows directly from the films into the groove at a rate which depends on its width. Since, however, this transfer coincides with copious oil transmission from the two normal sources, it constitutes a small proportion of the whole quantity transferred. The width of the collecting groove has therefore negligible effect on the rate of oil transfer, though it should obviously be sufficient to provide adequate oil distribution along the axial width of the bearing.

Oil Ring Guiding. A rotating oil ring is highly susceptible to actions, however small in themselves, which tend to displace the plane of the ring from its normal running position. Such disturbance results in an axial travel of the ring along the journal which may adversely affect ring operation. In many bearings, axial movement of the ring is controlled only by the sides of the oil ring gap in the bearing brasses; but experiment (Baildon 1937) has shown that control is most effectively exercised by the use of guides acting on the rising side of the ring and situated a short distance from the region of nearest approach.

Scraping Devices. The relatively meagre oil transfer by adhesive action at high journal speeds has encouraged the development of scooping and scraping devices which collect from the ring oil which is not available for adhesive transfer. Scoops (Karelitz 1930) are sometimes used to collect the oil exuded at the ring sides and also that contained in the side meniscal films in the region of journal and ring recession. Baudry and Tichvinsky (1937) describe the use of oil scrapers fitted to the lower half-bearing brass near its "exit" edge which collect oil from the sides of the rising portion of the ring and serve at the same time as ring guides.

The thick films which accumulate on the outer surface of the ring at high journal speeds provide an even more abundant source of oil supply. Difficulties in scraping are caused, however, by bodily shift of the ring due to progressive displacement from the vertical of the line of centres of journal and ring as journal speed increases. A scraper for collecting this oil should therefore be fixed vertically above the journal axis in such a position that the ring surface rises and almost touches the scraping edge when the journal attains its normal speed.

Special Ring Sections. The form of the inner surface of the ring controls the propulsive conditions and to a large extent the operation of adhesive oil transfer; hence in the design of this surface lie the most effective means of improving oil ring performance. In this respect, internal grooving of the ring, in the manner shown in Figs. 2 and 4,

prevents ejection of oil at the sides and ensures that thick oil films are retained in contact with the journal in the region where oil transfer takes place. The experimental results (Fig. 4) demonstrate the effects of such action in relatively profuse adhesive oil transfer at higher journal speeds and at all journal speeds when the oil viscosity is sufficiently high. With internal grooves the high intensity of loading on the propulsive film tends to establish boundary film propulsive conditions as indicated by curve 4 of Fig. 2. Such conditions result in relatively high ring speeds and, if scrapers are employed, appreciable increase of oil delivery is obtainable, due to both the higher rate of scraping and the thicker films which adhere to the ring.

Viscosity. Investigation of the effects of viscosity on both adhesive and scraping oil transfer shows that, except at very low journal speeds, oil transfer is less with oils of lower viscosity and therefore diminishes with increase in the temperature of any particular oil. Baudry and Tichvinsky (1937) have pointed out that this relationship introduces an important degree of instability into oil ring operation because an increase in bearing temperature results in reduced oil delivery and, in turn, diminished ability of the bearing to dissipate heat. Hence, further rise of temperature is induced and, if a temperature is reached at which oil delivery falls below the bearing requirements, ultimate seizure of the bearing is probable. These realities emphasize the necessity for due consideration of possible bearing temperature in the design of the ring oiling equipment for a bearing.

References

BAILDON, E. 1937 Thesis, Bradford Technical College, "Performance of Oil Rings, Chains, and Collars".

BAUDRY, R., and TICHVINSKY, L. M. 1937 *Mechanical Engineering*, vol. 59, p. 89.

CRANZ. 1912 Thesis, Verlagsanstalt, Stuttgart, "Versuche mit Schmierringen".

KARELITZ, G. B. 1930 Trans. A.S.M.E., vol. 52, Part I.

LASCHE, O. 1902 Z.V.D.I., vol. 46, p. 1934, "Die Reibungsverhältnisse in Lagern".

MÜLLER, K. 1932 Thesis, Technische Hochschule, Berlin, "Ölmengenmessungen an Ringschmierlagern".

THE LUBRICATION OF SYNTHETIC-RESIN BONDED BEARINGS

By P. Beuerlein*

Researches were undertaken to investigate the physico-chemical action of different groups of lubricants on a given synthetic-resin bonded material. The bushes used for the tests contained shreds of cotton fabric as filler and were known as type T. They were supplied ready for use by the H. Römmler A.-G. The measurements were: internal diameter, 18·10 mm.; external diameter, 24·05 mm.; length, 30 mm. As it had been found that the lubricant and cooling media had less effect on the untouched natural skin, while this skin wears away with normal use, part of the bushes was scraped externally to 23·85 and 23·45 mm. diameter. The following media were employed: pure mineral oils of different viscosities; compounded mineral oils; sulphurized mineral oils; soda soap grease; metallic soap grease; lime soap grease; emulsified greases; water of different values of hydrogen-ion concentration; and emulsions of different values of hydrogen-ion concentration.

Procedure. To avoid sources of error several bushes were tested simultaneously with each lubricant and cooling medium. With fluid media the bushes were soaked in the liquid and moved at definite intervals. In testing grease, the bushes were well covered with the grease and then placed in it. Where the tests concerned grease in presence of water, the bushes, after being given an even coat of grease, were placed in a water bath. All the bushes were measured at intervals of several days. The oils were first tested at room temperature, but as no changes in dimensions could be found, even after several weeks, the tests were repeated at 85 deg. C. All the other tests at room temperature revealed measurable changes and as a check they were also repeated under service temperatures. These results are not available at the time of writing. The results obtained so far are as follows:—

(1) *Greases.* For these tests, all the bushes were ground, and the following greases were employed:—

No. 1. Emulsified grease with a high water content, but no soap.

No. 2. Neutral soda soap grease; drop point, about 155 deg. C.; structure and consistency as for soft grease for roller bearings.

No. 3. Neutral soda soap grease blocks; drop point, about 180 deg. C.; consistency, very firm.

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* Rhenania-Ossag Mineralölwerke A.-G., Hamburg.

No. 4. Neutral soda soap grease; drop point, about 135 deg. C.; very soft, stringy grease made from an oil of unusually high viscosity.

No. 5. Lime soap grease; drop point, under 100 deg. C.; this was an ordinary machine grease (Stauffer grease) with added lime.

No. 6. Special lime soap grease; drop point, 100 deg. C.; tenacious and viscous consistency for particularly high pressures.

No. 7. Metallic soap grease, characterized by its content of oil of very high viscosity and its tenacious and viscous consistency.

The changes in dimensions (swelling) are recorded in Fig. 1, which refers to bushes which were covered with grease and placed in a water

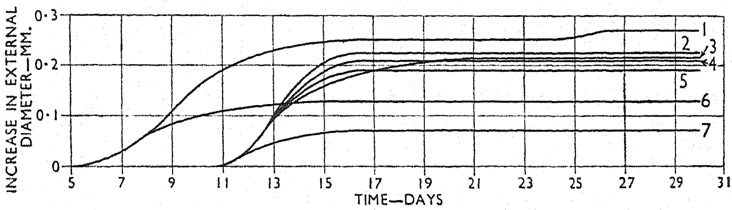


Fig. 1. Swelling due to Grease

The numbers refer to the greases tested.

bath. Most swelling was observed with grease No. 1. Greases 2, 3, and 4 fall into one group, No. 4 being the best; the high viscosity of the oil in No. 4 appears to act favourably. Greases 2, 3, and 4, without water, show swelling after several weeks' stay in the lubricant, but the swelling was 25 to 35 per cent less than that occurring (Fig. 1) in presence of water. Grease 5, which repels water, is similar to the best of the soda soap greases. Grease 6, which also repels water, causes less swelling. It is characteristic of greases 5 and 6 that when tested with or without water they show the same swelling; obviously the grease prevents water from reaching the surface of the bush, so that there is less swelling. Grease 7 achieves the lowest and best value and, like No. 6, is particularly suitable for highly loaded bearings cooled with water.

These tests show that for the preservation of such synthetic-resin bonded bearings, which in consequence of smaller load and higher velocity can be water-lubricated, it is advisable to give such bearings when stationary a small extra lubrication with greases 7 or 6 (cf. Fig. 2 for the values for pure water without the protective action of a water-repelling grease). No more grease must be used than is

required to protect the bush from the direct action of water. The tests show, further, that when cooling water is used, the bearings must be lubricated with a water-repelling grease. When emulsified greases are used, a larger clearance is required.

Synthetic-resin bonded bearings which are grease-lubricated but not water-cooled, show, owing to their low heat conductivity, higher temperatures in the lubricating film than do similar metallic brasses for the same frictional heat. Therefore, uncooled greased bearings of this material when in prolonged use should be lubricated with a soda soap grease of high drop point and heat resistance. The ordinary machine and grease-cup greases with drop points between 85–90 deg. C. are not sufficiently reliable under prolonged service. Greases of low drop point can, however, be used for synthetic-resin bonded bearings

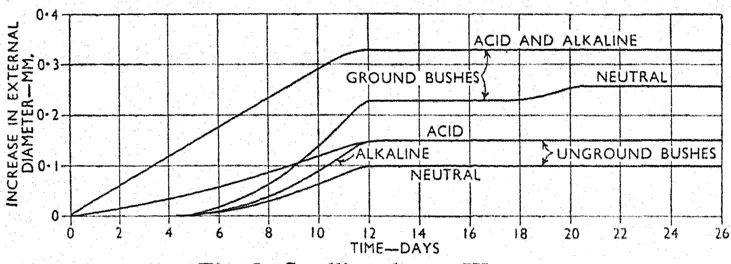


Fig. 2. Swelling due to Water

which are not in continuous service and have service intervals so that they can cool down.

(2) *Water.* Bushes with both ground and unground surfaces were tested with three types of water: neutral (hydrogen ion (pH) value, 7), acid (pH value, 1·4), and alkaline (pH value, 9·7). The results, recorded in Fig. 2, show that the swelling in neutral water was about 30 per cent less than in alkaline or acid water. Comparison with Fig. 1 shows, further, that the values for greases 2, 3, and 4 in the presence of water are roughly the same as for neutral water. The reason is that these greases do not repel water. Comparison with the water-repelling greases 6 and 7 (Fig. 1) clearly shows the excellent protection afforded by suitable lubricants against the action of pure water, though it should be remembered that the cooling water is rarely neutral, being usually alkaline and less frequently acid. The tests with unground bushes revealed similar differences, except that the swelling, as would be expected, was smaller.

(3) *Emulsions.* Except for a few preliminary tests with unground

bushes, these tests were carried out on ground bushes. The problem was to determine whether emulsions acted differently from pure water in respect of swelling and also whether the presence of a small amount of alkali in the emulsion had any effect on swelling power. Preliminary tests with bushes soaked for five weeks in emulsions made from an ordinary drilling oil and water mixed in the proportions 1/10, 1/20, and 1/50 showed that in general the swellings were the same as in water. It was noteworthy that with these emulsions the unground bushes swelled more than the ground bushes. This indicates that the action of emulsions is liable to be unpredictable.

Preliminary tests were also made with emulsions prepared from a highly emulsifiable corrosion-preventive oil and the same water as before, using the same proportions of oil to water. This oil yields an unusually fine emulsion. On the average the swelling with these emulsions was somewhat less than with pure water, whilst the maximum swelling was the same as with pure water. This shows that emulsifiable oils do not all act similarly. All the emulsions used in these preliminary tests were slightly alkaline.

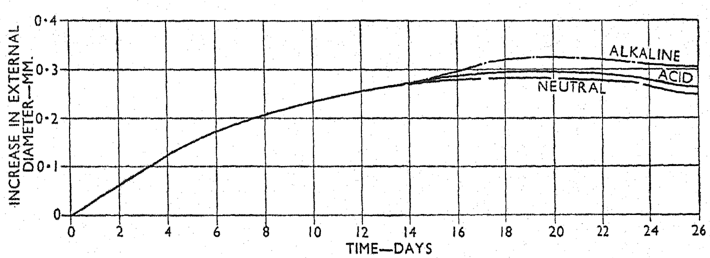


Fig. 3. Swelling due to Emulsions

Next, three 1/10 emulsions were prepared with the corrosion-preventive oil and made up to be acid, alkaline, and neutral, the *p*H values being 6·45, 7·75, and 7 respectively. The values found (Fig. 3) lay very close together. The neutral emulsion produced more swelling than did neutral water, whereas the alkaline and acid emulsions showed but slight differences compared with water. It is practically impossible to prepare a neutral emulsion from an emulsifiable oil with any type of water; usually the emulsions have *p*H values over 7, and rarely under 7. For this reason an emulsion prepared in the works will cause greater swelling than a neutral water, whereas it will hardly cause more swelling than an alkaline or acid water. It is therefore difficult to lay down uniform directions for lubrication with emulsions.

When rolling-mill bearings lubricated with water are stopped for

short periods they should be treated with some emulsifying corrosion-preventive oil to prevent rusting and to assist starting-up. The curves show that such corrosion-preventive oils do not cause any greater swelling than the water itself. The best protection for synthetic-resin bonded bearings is given by greases 6 or 7.

(4) *Oils.* Twelve oils were tested with ground and unground bushes:—

TABLE 1. TYPES OF OILS TESTED

Oil	Type of oil	Approximate Engler viscosity at 50 deg. C.
1	Compounded mineral oil	8·5
2	,, ,, ,,	11·5
3	Sulphurized mineral oil	19
4	Pure mineral oil	11·5
5	,, ,, ,,	9·5
6	,, ,, ,,	4·5
7	,, ,, ,,	2·8
8	Pure cylinder oil	55 (6·3 at 100 deg. C.)
9	Compounded mineral oil	4·5
10	Sulphurized mineral oil	55
11	Compounded cylinder oil	40 (5·0 at 100 deg. C.)
12	Compounded mineral oil	2·5

As the material showed no change after soaking in oil for four weeks at room temperature, the temperature of the oil bath was raised to 85 deg. C. The bushes were measured at about 80 deg. C. In contrast with the first three groups, group 4 revealed no swelling, but actually shrank. The curves for bushes ground externally are reproduced in Fig. 4, and those for unground bushes in Fig. 5. The inner and outer diameters were found to decrease in the same proportion. No further change took place on cooling to air temperature. There was no change on carrying out the first measurement (after 17 hours), probably owing to the poor heat conductivity. Owing to the dispersion of the results,* it was impossible to classify the oils according to their characteristics. The sulphurized oils behaved similarly to the unsulphurized oils. The compounded mineral oils give more scattered results than the pure mineral oils, though the averages correspond. Oil 1 gives the most shrinkage and oil 12 (Table 1) the least shrinkage (Fig. 4), both being compounded mineral oils. At equal

* In these tests only one ground and one unground bush could be placed in the oil bath.

viscosities for Nos. 1 and 5, and also 2 and 4, the pure mineral oil is better, and for Nos. 6 and 9, and also 7 and 12, the compounded oil. It is to be noted that all the oils with an Engler viscosity of over 8·5 at 50 deg. C. produce the final shrinkage irregularly after 17 and before

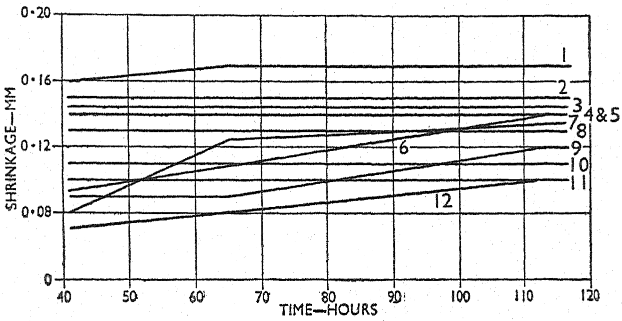


Fig. 4. Shrinkage due to Oils (Ground Bush)

The numbers refer to the oils tested.

41 hours, whereas the more fluid oils (6, 7, 9, 12) produce a more gradual shrinkage, which also begins only after 17 hours. This point requires attention with bushes which are to be subject to high temperatures. Comparison of Figs. 4 and 5 reveals that unground bushes shrink somewhat less than the ground bushes, the order being different.

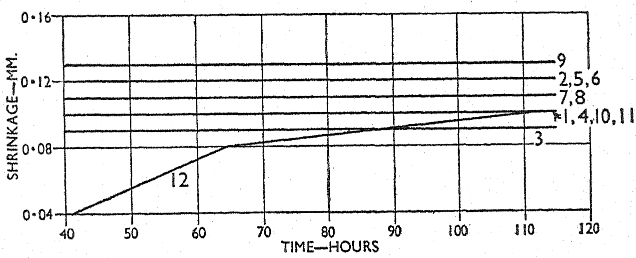


Fig. 5. Shrinkage due to Oils (Unground Bush)

The numbers refer to the oils tested.

The order in Fig. 5 agrees with the dispersion shown in Fig. 4. The difference in shrinkage is not great; the values, except for oil No. 1, lie between 0·1 and 0·15 mm. in Fig. 4, and between 0·09 and 0·13 mm. in Fig. 5. As, apart from the shrinkage, no swelling of the surfaces could be found, the slightly greater shrinkage of the ground bushes must be due to their thinner walls and the consequent loss of strength. Further tests will be made to find the influence of wall

thickness on shrinkage and the effect of the fillers used in synthetic-resin bonded bushes.

Bearing Clearance and Load Capacity. The directions for the preparation and use of bearings and bushes made from synthetic resin compositions, published by the "Fachausschuss für Kunst- und Pressstoffe" of the Verein Deutscher Ingenieure, recommend upper limits for the wide, smooth running fits for such bearings. As the swelling due to the lubricant proceeds but slowly, a considerable time will elapse before the clearance becomes smaller. According to the hydrodynamic theory, the load capacity of a bearing depends upon the clearance, so that it is important to know how far the loss of load capacity due to the change in clearance can be compensated by choosing a suitable lubricant. This was found by means of formulæ due to Falz*, the calculation being made by Dr.-Ing. F. Hamacher. Calculations were made for synthetic-resin composition bearings with the same least permissible film thickness as for metal bearings and for the same temperature in the layer of lubricant. The clearance was reckoned according to DIN (Deutsche Industrie-Normen) standards as well as to the gauge units (*Passungs-Einheiten*=PE) as these are simpler to handle than the ISA-*Passungen*. The notation used is as follows:—

1PE	$= 0{\cdot}05\times10^{-3}\sqrt[3]{d}$
D	Diameter of the bore.
d	Diameter of the shaft.
ω	Angular velocity.
$(1-\chi)$	Relative film thickness.
z	Viscosity of the lubricant.
E	Engler degrees.
n	Shaft revolutions per minute.
p_m	Average load.
$p_{m_{\max}}$	Maximum permissible average bearing pressure.
$h_{\min}$	Least permissible film thickness.
γ	Specific gravity of the lubricant (taken as 0·9).
ψ	$= \sqrt{0{\cdot}52/(1-\chi)}\cdot\sqrt{z\omega/p_m}$
q	$= \sqrt[3]{d}/(\mathrm{D}-d)$
$h_{\min}$	$= \ldots.(1-\chi)\sqrt[3]{d}/2q = 0{\cdot}26\times q\times z\omega/p_m\sqrt[3]{d^5}$
$p_{m\,\max}$	$= 0{\cdot}26\times q\times z\omega/h_{\min}\sqrt{d^5} = 0{\cdot}0272\times q\times nz/h_{\min}d^{1{\cdot}67}$.

Nowadays, synthetic resin bearings replace metal bearings mainly where very close fitting is not required. Therefore the calculation is

* Falz, E. "Grundzüge der Schmiertechnik", Springer, Berlin, 1931.

limited to easy, smooth, wide easy, and wide smooth-running fits. The average values of these fits were taken and the following calculations made:—

Running fits	Average value in gauge units (PE)	$q \times 10^{-3}$
Easy	4·75	4·22
Smooth	4·75	4·22
Wide, easy	6·75	2·97
Wide, smooth . . .	9·25	2·17
Upper limit of the wide, smooth-running fits	13·5	1·49

The expression z is calculated in Engler degrees according to the Ubbelohde formula $z=0{\cdot}000666E-0{\cdot}00064/E$. With this formula the increase in viscosity has been calculated which is required to obtain the same $p_{m\ \max}$ and thus the same load capacity when changing from a small clearance (wide-running fit) to the upper limit of fit of the wide smooth-running fit. For a smaller clearance the viscosity required

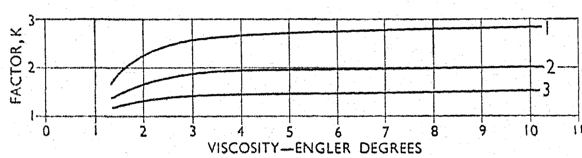


Fig. 6. Oil Viscosity at Service Temperature for the Smaller Clearance

Curve 1 Easy or smooth-running fit (average value).
,, 2 Wide, easy-running fit (average value).
,, 3 Wide, smooth-running fit (average value).

is multiplied by the factor k so as to obtain the same load capacity as with the larger clearance of the wide smooth-running bearing. This factor k is expressed graphically in Fig. 6. In using these curves it must be considered that for bearing lubrication the viscosity is always generously high to ensure safety and in changing from a metal bearing with a smaller clearance to a composition bearing, application of the factor k generally gives too high a viscosity. It is therefore advisable to ascertain the viscosity which is *really* necessary for the metal bearing to carry the load. Using this viscosity, the calculation should be made for the larger clearance of the synthetic-resin bearing. There is no fear that these bearings may prove less reliable in service, as there is no danger of harm to the journal from sudden overloads, dust particles,

etc., with them. Though the curves show that it is expedient to use oils of somewhat higher viscosity, it is not advisable to go too far in the matter as the thinner oils have better cooling capacity, and a compromise must be effected. Further, before fitting the brasses, they should be soaked in warm oil for long enough to reduce the clearance at the start.

CHARACTERISTICS OF THE 120 DEG. JOURNAL BEARING

By R. O. Boswall, D.Sc. (Eng.), M.Sc.Tech.*

The following notation is adopted in this paper :—

D Diameter of the bearing.
R Radius of the bearing.
B Length of the bearing.
r Radial clearance or difference between the radius of bearing surface and journal.
ψ Angle subtended by the bearing surface.
δ Angular position of the line of action of the load measured from the inlet edge of the bearing surface.
W Total load transmitted by the journal.
N Speed of journal rotation, revolutions per minute.
Z Viscosity of the lubricant supplied to the bearing, measured in poises.

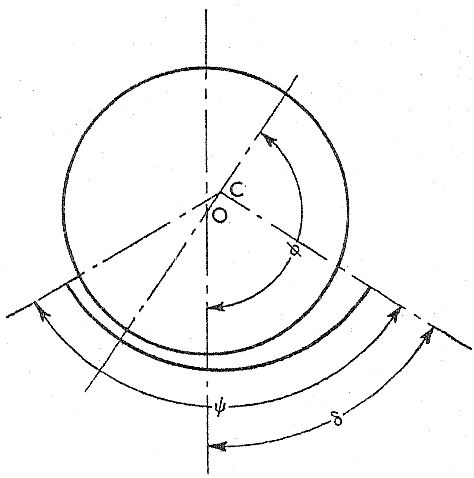


Fig. 1. Attitude of the Journal

Journal Attitude and Film Thickness. The attitude of the journal relative to the fixed bearing surface (Fig. 1) is given by :—

(*a*) The distance OC between the centres of the journal and bearing surface. This will be denoted by e.

* College of Technology, Manchester.

(*b*) The angle ϕ between the line joining these centres and the line of action of the load.

Writing $P=W/BD$, the distance e and the angle ϕ can be expressed by

$$(e/r)=C_a(ZN/P)^{\frac{1}{2}}(R/r)$$
$$\phi=C_b(ZN/P)^{\frac{1}{2}}(R/r)$$

where C_a and C_b are non-dimensional coefficients dependent upon the angular dimensions of the bearing surface, the ratio between length and diameter, and the change in viscosity that takes place owing to temperature changes in the lubricant as it flows through the clearance space between the journal and bearing surface.

The film thickness at any point is calculated from $h=(r+e\cos\theta)$, where θ is the angle between the line OC and the radius to the point where the film thickness is measured.

Coefficient of Friction. The coefficient of friction is expressed by

$$\mu(R/r)=C_d(ZN/P)^{\frac{1}{2}}(R/r)$$

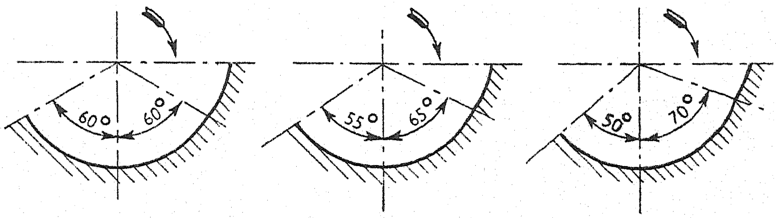
Quantity of Lubricant. The volume of lubricant required per second for complete formation of the film will be denoted by Q and is given by

$$Q(R/r)=C_e(NBD^2)(ZN/P)^{\frac{1}{2}}(R/r)$$

Conditions for a Bearing Surface Subtending 120 *deg.* Approximate values for (e/r), ϕ, $\mu(R/r)$, and $Q(R/r)(NBD^2)$ corresponding to various values for the criterion $(ZN/P)(R/r)^2$ have been calculated for a bearing surface extending over an angle of 120 deg., the assumption being made that, owing to increase in temperature, the viscosity of the lubricant leaving the bearing will be half the viscosity of the lubricant supplied to the bearing. These values, given in Tables 1, 2, and 3, enable the operating conditions to be determined with an accuracy sufficient for most practical purposes for three different ratios of length to diameter and three positions for the line of action of the load.

An important feature of the film-lubricated journal bearing is the possibility of negative pressure occurring near the outlet end of the bearing and it may be assumed that, provided the value for $(ZN/P)(R/r)^2$ does not fall below the minimum value given in the tables, negative pressure conditions will not occur and the film should form effectively over the complete angle subtended by the bearing surface. It will be noticed that if the value for the criterion $(ZN/P)(R/r)^2$ is less than about 10, effective film formation can only be obtained by using a relatively wide bearing with offset loading.

Dimensions. Values for (ZN/P) given in the tables are in metric units, i.e. Z is in poises, P is in dynes per square centimetre, and N is in revolutions per minute. Values for P expressed in pounds per square inch can be converted into dynes per square centimetre by multiplying by 69,000.



a *b* *c*

Fig. 2. (*a*) Central Loading; (*b*) Offset Loading; (*c*) Offset Loading

TABLE 1. CENTRAL LOADING AS SHOWN IN FIG. 2 *a*

Length-diameter ratio	$(ZN/P)(R/r)^2$	e/r	ϕ, deg.	$\mu(R/r)$	$Q(R/r)/(NBD^2)$
2·0	9·0	0·625	146·5	1·55	0·0072
	10·0	0·600	146·0	1·62	0·0075
	12·5	0·535	144·0	1·78	0·0082
	15·0	0·480	142·0	1·95	0·0088
	17·5	0·440	140·0	2·10	0·0093
	20·0	0·410	138·0	2·27	0·0098
	25·0	0·350	134·0	2·60	0·0107
	30·0	0·300	130·5	2·93	0·0115
1·5	13·5	0·555	148·5	2·05	0·0080
	15·0	0·525	148·0	2·15	0·0082
	17·5	0·490	147·0	2·32	0·0086
	20·0	0·455	146·5	2·50	0·0090
	25·0	0·400	144·5	2·85	0·0097
	30·0	0·350	140·5	3·20	0·0104
1·0	25·0	0·500	151·0	3·37	0·0093
	30·0	0·460	151·0	3·70	0·0098
	35·0	0·420	150·5	4·07	0·0100
	40·0	0·390	149·5	4·40	0·0103

The value for the displacement e of the journal centre must be in the same units as the radial clearance r.

The volume Q will be expressed in cubic centimetres per second if B and D are measured in centimetres, or in cubic inches per second if they are measured in inches. The conversion of cubic centimetres and cubic inches per second to gallons per minute is effected by multiplying these quantities by 0·01324 and 0·217 respectively.

TABLE 2. OFFSET LOADING AS SHOWN IN FIG. 2 *b*

Length-diameter ratio	$(ZN/P)(R/r)^2$	e/r	ϕ, deg.	$\mu(R/r)$	$Q(R/r)/(NBD^2)$
2·0	5·0	0·720	148·0	1·10	0·0060
	7·5	0·640	145·0	1·30	0·0073
	10·0	0·575	141·0	1·47	0·0084
	12·5	0·525	136·5	1·63	0·0092
	15·0	0·475	131·5	1·78	0·0100
	17·5	0·435	127·0	1·93	0·0106
	20·0	0·400	122·0	2·07	0·0112
	25·0	0·340	112·5	2·37	0·0123
	30·0	0·300	104·0	2·66	0·0133
	40·0	0·240	89·0	3·25	0·0150
1·5	7·0	0·690	149·0	1·38	0·0068
	10·0	0·605	146·0	1·60	0·0081
	12·5	0·550	143·0	1·78	0·0090
	15·0	0·500	140·0	1·95	0·0096
	17·5	0·460	137·0	2·10	0·0102
	20·0	0·425	134·0	2·26	0·0108
	25·0	0·370	128·0	2·58	0·0116
	30·0	0·325	122·5	2·90	0·0122
	40·0	0·280	113·0	3·55	0·0129
1·0	14·0	0·630	151·0	2·35	0·0083
	15·0	0·615	150·5	2·41	0·0085
	17·5	0·575	149·0	2·59	0·0090
	20·0	0·530	148·0	2·75	0·0095
	25·0	0·465	145·0	3·07	0·0098
	30·0	0·415	142·0	3·40	0·0108
	40·0	0·340	136·0	4·02	0·0118

Method of Using the Data. The application of the given data to determine the approximate operating conditions for a bearing having a diameter of 10 inches and a length of 15 inches is shown in Table 4. The viscosity of the lubricant supplied to the bearing has been taken as 0·5 poise and, owing to the assumed viscosity change, the viscosity of the lubricant when it leaves the bearing will be 0·25 poise.

It will be seen that with offset loading the film forms over the complete

angle of 120 deg. without risk of interference from negative pressure conditions at a lower speed than with central loading. For example with a central load of 10,000 lb. and (R/r) equal to 250, the film will not have completely formed until the speed reaches 2,000 r.p.m., whereas with offset loading at 65 deg. from the inlet edge the complete film will probably have formed before the speed has reached 1,000 r.p.m. Offset loading slightly increases the film thickness and very noticeably reduces the coefficient of friction. Increase of the clearance ratio (R/r) seems to encourage the effective formation of the film at lower speeds.

TABLE 3. OFFSET LOADING AS SHOWN IN FIG. 2 *c*

Length-diameter ratio	$(ZN/P)(R/r)^2$	e/r	ϕ, deg.	$\mu(R/r)$	$Q(R/r)/(NBD^2)$
2·0	3·5	0·775	147·5	0·90	0·0054
	5·0	0·710	144·0	1·02	0·0067
	7·5	0·625	136·5	1·20	0·0083
	10·0	0·570	128·5	1·37	0·0097
	12·5	0·530	120·0	1·51	0·0108
	15·0	0·510	112·0	1·65	0·0118
	17·5	0·500	103·5	1·78	0·0127
	20·0	0·495	96·5	1·90	0·0135
	25·0	0·500	84·0	2·14	0·0148
	30·0	0·510	74·0	2·35	0·0157
	40·0	0·520	62·0	2·80	0·0170
1·5	4·0	0·765	149·5	1·03	0·0058
	5·0	0·720	147·5	1·12	0·0067
	7·5	0·655	142·5	1·31	0·0082
	10·0	0·595	137·5	1·48	0·0094
	12·5	0·545	132·0	1·65	0·0102
	15·0	0·505	126·5	1·82	0·0108
	17·5	0·475	121·0	1·97	0·0113
	20·0	0·455	116·0	2·11	0·0117
	25·0	0·430	107·0	2·42	0·0124
	30·0	0·410	99·5	2·72	0·0130
	40·0	0·390	86·0	3·30	0·0140
1·0	8·0	0·725	152·0	1·60	0·0073
	10·0	0·685	150·5	1·76	0·0081
	12·5	0·640	147·5	1·97	0·0090
	15·0	0·595	144·5	2·15	0·0098
	17·5	0·550	141·0	2·35	0·0105
	20·0	0·520	138·0	2·50	0·0112
	25·0	0·460	131·5	2·84	0·0122
	30·0	0·415	125·5	3·16	0·0128
	40·0	0·350	114·0	3·82	0·0138

TABLE 4

W, lb.	N, r.p.m.	(R/r)	(ZN/P)(R/r)²	e, inches	ϕ, deg.	h_m, inches	μ	Q, gal. per min.
(a) Central Loading:—								
10,000	2,000	250	13·6	0·0112	148·5	0·0088	0·0082	20·8
	3,000	250	20·4	0·0090	146·5	0·0110	0·0101	35·0
10,000	1,000	500	27·2	0·0038	143·0	0·0062	0·0060	6·5
20,000	1,000	500	13·6	0·0056	148·5	0·0044	0·0041	5·2
	2,000	500	27·2	0·0038	143·0	0·0062	0·0060	13·0
	3,000	500	40·8	0·0027	127·0	0·0073	0·0079	23·4
30,000	2,000	500	18·1	0·0048	147·0	0·0052	0·0048	11·5
	3,000	500	27·2	0·0038	143·0	0·0062	0·0060	19·5
(b) Offset Loading. Load line 65 deg. from Inlet Edge:—								
10,000	1,000	250	6·8	0·0139	149·0	0·0061	0·0055	8·8
	2,000	250	13·6	0·0105	142·0	0·0095	0·0074	24·1
	3,000	250	20·4	0·0084	133·5	0·0116	0·0092	42·1
20,000	2,000	250	6·8	0·0139	149·0	0·0061	0·0055	17·6
	3,000	250	10·2	0·0120	145·5	0·0080	0·0065	32·0
30,000	3,000	250	6·8	0·0139	149·0	0·0061	0·0055	26·4
10,000	1,000	500	27·2	0·0035	124·5	0·0065	0·0054	7·7
20,000	1,000	500	13·6	0·0052	142·0	0·0048	0·0037	6·0
	2,000	500	27·2	0·0035	125·5	0·0065	0·0054	15·4
	3,000	500	40·8	0·0028	112·0	0·0073	0·0072	25·4
30,000	1,000	500	9·1	0·0063	147·0	0·0037	0·0031	5·1
	2,000	500	18·1	0·0045	136·5	0·0055	0·0043	13·5
	3,000	500	27·2	0·0035	125·5	0·0065	0·0054	23·0
(c) Offset Loading. Load line 70 deg. from Inlet Edge:—								
10,000	1,000	250	6·8	0·0135	144·0	0·0065	0·0051	10·1
	2,000	250	13·6	0·0105	130·0	0·0095	0·0069	27·3
	3,000	250	20·4	0·0092	115·0	0·0111	0·0086	45·6
20,000	2,000	250	6·8	0·0135	144·0	0·0065	0·0051	20·3
	3,000	250	10·2	0·0117	137·0	0·0083	0·0060	37·0
30,000	3,000	250	6·8	0·0135	144·0	0·0065	0·0051	30·4
10,000	1,000	500	27·2	0·0042	103·0	0·0063	0·0051	8·3
20,000	1,000	500	13·6	0·0052	130·0	0·0048	0·0035	6·8
	2,000	500	27·2	0·0042	103·0	0·0063	0·0051	16·5
	3,000	500	40·8	0·0038	92·5	0·0070	0·0067	27·3
30,000	1,000	500	9·1	0·0061	139·5	0·0039	0·0028	5·9
	2,000	500	18·1	0·0048	120·0	0·0053	0·0040	14·8
	3,000	500	27·2	0·0042	103·0	0·0063	0·0051	24·8

RELATIONSHIP OF THE PRESSURE-VISCOSITY EFFECT TO BEARING PERFORMANCE

By Professor Louis J. Bradford and C. G. Vandegrift *

In the bearing operating under conditions which permit the complete separation of the rubbing surfaces the only quality of the lubricant to enter the picture is the viscosity. Once the geometrical proportions of the bearing and the rubbing speed of the surfaces are fixed, the capacity and friction of the combination are determined by the viscosity of the fluid constituting the separating film. By this is meant the viscosity of the lubricant as it exists in the clearance space and under conditions of operation.

The viscosity of the lubricant in the film differs from that supplied to the leading edge of the bearing because of the heat due to friction, and the pressure imposed by the load on the bearing. The first of these causes a sharp decrease in the viscosity, and has long been allowed for. The second causes an almost equally sharp rise in the viscosity. It has not been known as long as has the heat effect, and no allowance for it has been made by engineers generally, because its influence is not marked at the bearing pressures commonly employed. The increase of viscosity with pressure is, nevertheless, an important quality which, if properly used, may enable the production of bearings of many times the capacity of present-day bearings, and, moreover, bearings that will not wear out. Fig. 1 shows the relationship of viscosity to pressure for three oils, as determined by Dow (1936 †). It will be noted that while there is a marked difference between the several oils, the viscosity at the higher pressure is always many times that of the same oil at atmospheric conditions.

In the film separating the rubbing surfaces of a loaded bearing the pressure varies from zero at the leading edge, through a maximum and then back to zero at the trailing edge. This pressure depends, among other things, upon the viscosity of the oil within the film, and the viscosity is, in turn, affected by the pressure. In general, the higher the viscosity the higher the pressure, and the higher the pressure the higher the viscosity, other conditions being equal.

In the mathematical work on the behaviour of bearings operating under fluid film conditions this variation of viscosity with pressure has been neglected with justification, because the bearing pressures used in practice have been comparatively low. If, however, the bearing

* Pennsylvania State College.
† Society of Rheology, October meeting, 1936.

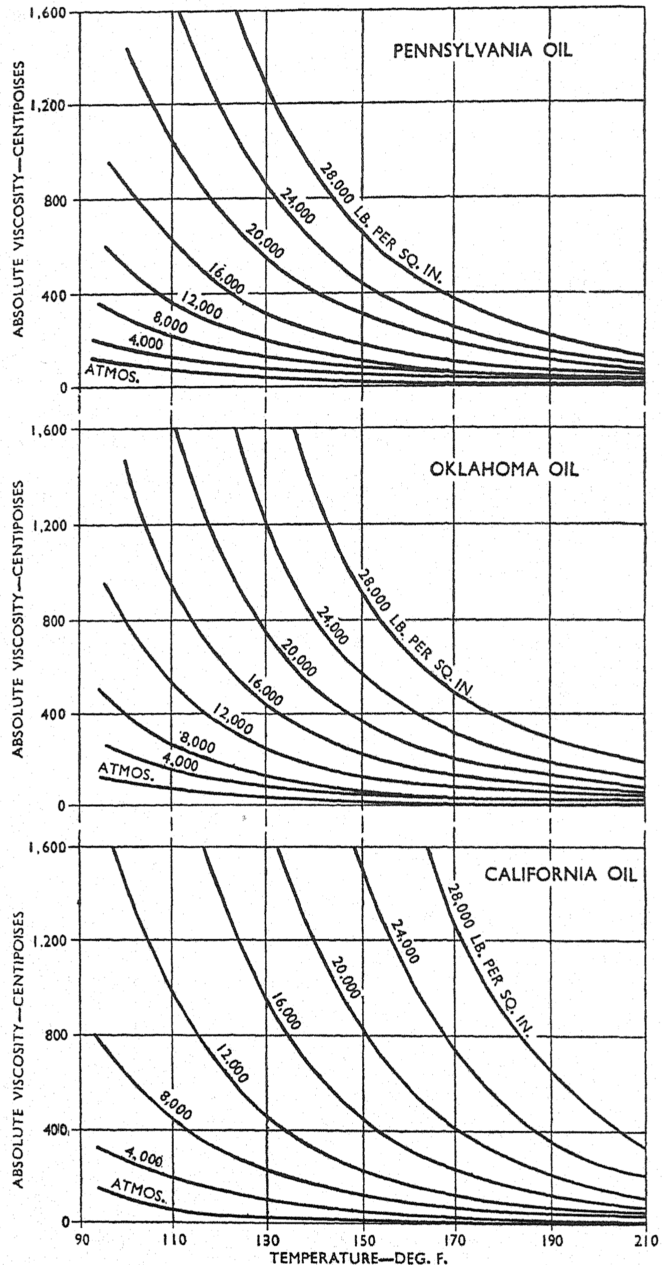


Fig. 1. Relation of Viscosity to Pressure for Three Different Oils

pressures are greatly raised the effect of pressure on viscosity cannot be neglected, and the mathematics involved must be arranged to take account of it.

Taking the hydrodynamic equations of Lamb, using the viscosity as a variable, and combining these with the equations of motion, equations are obtained which define the fundamental relationship of the quantities involved, that is, the dimensions of the channel, the relative speed of the surfaces, the pressure within the film, and the viscosity of the lubricant. The last of these can be written as a function of the pressure. For a flat plate of infinite length at right-

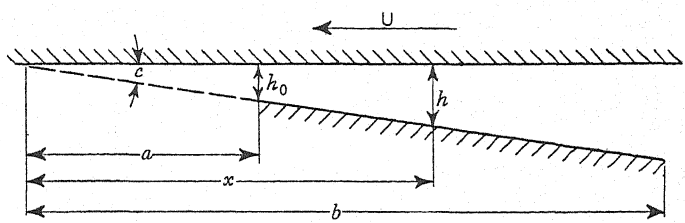


Fig. 2. Definition of Symbols and Values of Constants

a 0·0818 inch; b=0·2068 inch; $c=h_0/a$.
μ Coefficient of viscosity, variable.
μ_0 Coefficient of viscosity of entering oil.
c Small angle of inclination of plates.
p Pressure in the film at any value x.
h Film thickness at any value x.
W Load per unit of width.
α Slope of pressure versus log μ/μ_0 curve.
U Velocity of the moving surface, 47·1 in. per sec.

angles to the direction of motion the viscosity of the lubricant at any point within the film is given by the expression

$$\mu = \frac{\mu_0}{\dfrac{6\mu_0 U}{c^2\alpha}\left[\dfrac{(x-a)(x-b)}{x^2(a+b)}\right]+1}$$

(for the notation, see Fig. 2). The pressure corresponding to μ can be obtained from the pressure-viscosity test data, and this value is then plotted on its position within the bearing. This yields a curve of pressures the area under which represents the carrying capacity of the bearing under the assumed conditions of size, film thickness, lubricant, speed, and inclination of the surfaces.

Curve a (Fig. 3) shows the pressure distribution with a certain bearing when the variation of viscosity with pressure is considered, while curve b shows the pressure distribution when calculated for a constant

viscosity equal to that of the lubricant at atmospheric pressure. The inclination of the plates to each other and the minimum film thickness are the same in each case. The difference in the bearing capacities is striking, and suggests the possibility of very high-capacity bearings which still operate in the fluid film region. Fig. 5 illustrates the effect of film thickness on capacity. With comparatively thick films the pressure-viscosity effect is small but, as the film thickness is reduced, the effect becomes marked, and, with this particular bearing,

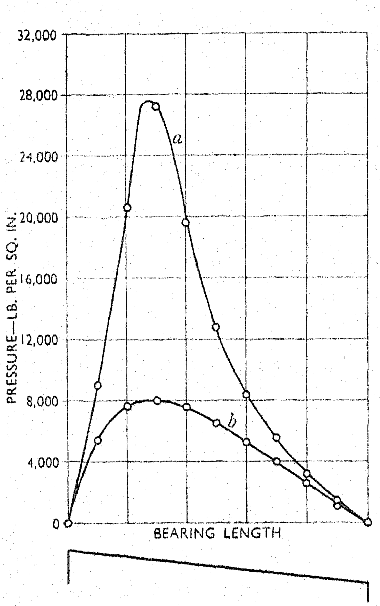


Fig. 3. Pressure Distribution in a Bearing

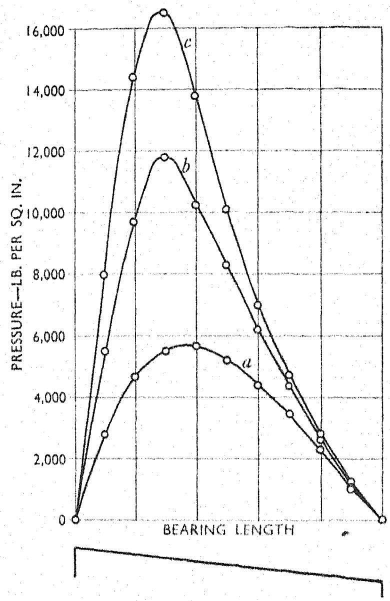


Fig. 4. Pressure Curves in Relation to Frictional Heating

the maximum pressure, and hence the capacity of the bearing, becomes infinite for a minimum film thickness of 0·0000295 inch.

The gains in capacity indicated by Figs. 3 and 5 cannot, however, be wholly realized, because the heating due to friction causes a reduction in the viscosity of the lubricant, and this offsets the gain due to pressure. Fig. 4 reproduces the pressure curves obtained when this heating is taken into consideration. Curve *a* assumes a uniform temperature rise from 140 to 180 deg. F. as the lubricant passes through the bearing, while curve *b* shows the results when the change is from 140 to 150 deg. F. Curve *c* considers no heating effect. The reduc-

tion in capacity caused by heating is marked. These curves are drawn for a minimum film thickness of 0·0000316 inch.

The theoretical investigations outlined briefly above carry some interesting indications for designers and manufacturers of bearings and also for producers of lubricants. It will be noted that, for the bearing considered, a minimum film thickness of 0·0000295 inch represented the limiting thickness of the film. Provided this dimension could be reached without the occurrence of metallic contact, and without the temperature of the film rising, the load that could be imposed would be limited only by the strength of the bearing and journal. Smoothness of bearing and journal are, therefore, essential.

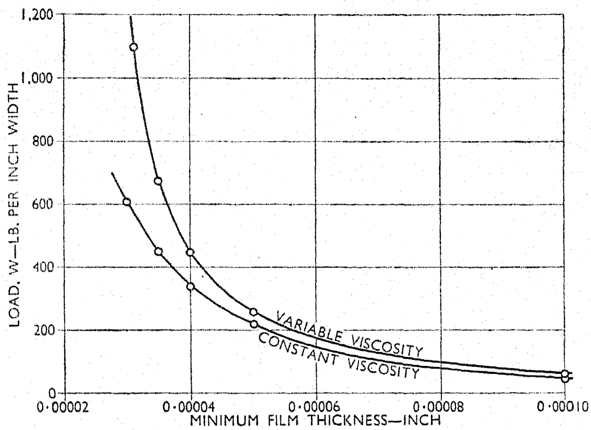


Fig. 5. Effect of Film Thickness on Bearing Capacity

The curves of Fig. 4 emphasize the necessity of abstracting the heat caused by friction as soon as it is formed.

The very minute thickness of the film may suggest that it is impossible to obtain rubbing surfaces of sufficient smoothness to permit exploitation of the pressure-viscosity effect. This is doubtless true to-day, but it is by no means certain that it will always be true. Indeed an experimental bearing of the size used in computing the curves referred to above has been in use in the laboratories of the Pennsylvania State College for over two years and has repeatedly sustained loads of 24,000 lb. per sq. in. for many hours without showing any sign of wear or seizure. It is believed that modern industry can meet the exacting requirements of machining and alignment if convinced that the result is commercially profitable.

The effect upon bearing capacity of the reduction in viscosity caused

by heating can be minimized by using a lubricant with a high viscosity index. In fact the ideal lubricant, judged by its ability to prevent metallic contact between the rubbing surfaces, is one whose viscosity decreases but little with increasing temperature, but increases rapidly with pressure. Such a combination is the opposite to that found in the oils commercially available, but may possibly be secured by altered refining methods.

Friction is of importance second only to capacity. In Fig. 6 the coefficient of friction for the bearing previously cited is plotted against the film thickness. Curve *a* gives the values obtained if the viscosity is kept constant at the value obtaining at atmospheric pressure, while curve *b* gives the values obtained when the effect of pressure is considered. This indicates that rather high friction losses may be expected in bearings of extreme capacity. It also suggests that oils

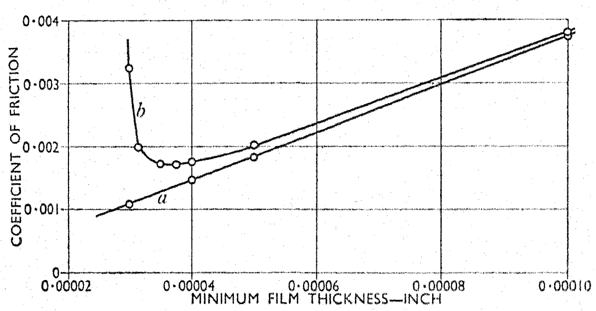


Fig. 6. Coefficient of Friction in Relation to Film Thickness

having different pressure-viscosity effects will yield different coefficients of friction in the high-pressure region, even though the viscosities at atmospheric pressure are identical. This in turn suggests that a large part of the difference in friction coefficients of oils now attributed to that elusive property "oiliness" should properly be attributed to the difference in the variation of viscosity with pressure.

W. H. Herschel * has defined "oiliness" as follows: "If two oils having the same viscosity at the running temperature of the film give different coefficients of friction when tested under identical conditions, the one giving the lower coefficient is said to be the more oily." Judged by this standard, lard oil is universally conceded to be more oily than mineral oil. Fig. 7 shows the coefficients of friction to be expected with the bearing previously used, when supplied with lard oil and with Pennsylvania mineral oil of equal viscosity at the assumed temperature

* Jl. Soc. Automotive Eng., 1922, vol. 10, p. 31.

and atmospheric pressure, and the effect of pressure considered. These values have been plotted against the load on the bearing. The conditions of the Herschel definition have been met and the curves show that: (1) the two oils give different coefficients for the same load; (2) the coefficient for lard oil is the lower; (3) the differences are greatest in the high-pressure region. Had these values been obtained by experiment we should certainly have concluded that lard oil was more "oily" than was mineral oil. The difference is, however, in this case due solely to the pressure-viscosity effect and not to any mysterious property called "oiliness".

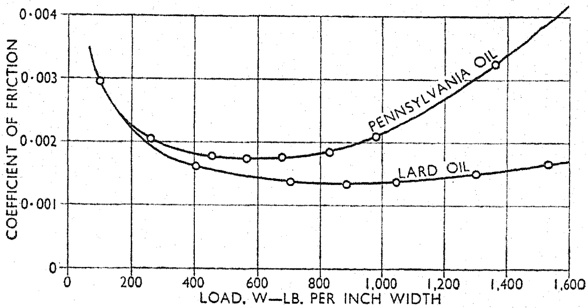


Fig. 7. Effect of Pressure on the Coefficients of Friction for Lard Oil and Pennsylvania Oil

In conclusion it may be said that the pressure-viscosity relationship appears to be fully as important as the temperature-viscosity relationship, and that it holds out the possibility of bearings of extremely high capacities that still operate under fluid conditions. To make this possible, both bearing and journal must be very smoothly finished and so mounted as to permit the formation of a film that is convergent in the direction of motion, and in addition the bearing should be cooled in such a way as to remove the heat of friction as soon as it is formed in the bearing. A lubricant for a high-capacity bearing should have as large a pressure-viscosity effect as possible combined with a small temperature-viscosity effect. Such a lubricant will, however, give higher friction for a given load than one whose viscosity rises less sharply with increasing pressure.

RUBBER AS A MATERIAL FOR BEARINGS

By Sidney A. Brazier, M.Sc.,* and W. Holland-Bowyer *

In comparison with many other materials, vulcanized rubber, when dry, has a high coefficient of friction. In view of the many uses to which rubber is put because of this, it may seem somewhat surprising that it has also been found eminently suitable for use as a bearing material where relatively low frictional resistance is required. This is due to the fact that the coefficient of friction is considerably reduced when the contacting surfaces are water-lubricated, under which condition it may actually become less than that of many oil-lubricated metal bearings. The following coefficients of running friction (for low pressures) for typical materials, taken from various sources, will serve to illustrate these points:—

Bronze on bronze (dry), 0·2
,, ,, ,, (wet and clean), 0·3
Rubber on iron (dry), 0·5 to 1·14
Steel on steel (slightly greasy), 0·15
,, ,, (greasy), 0·07 to 0·08
,, ,, (lubricant continually renewed), 0·04 to 0·05

The following figures are quoted from data published by Busse and Denton (1932):—

Water-lubricated bearings	Bearing load, lb. per sq. in.	Coefficient of friction
Water-lubricated metal bearing, steel shaft	2	0·02
,, ,, ,, ,,	35	0·25
Fluted rubber bearing, steel shaft . .	2	0·05
,, ,, ,, ,, . .	35	0·05
Spiral ,, ,, ,, . .	150	0·038
,, ,, ,, ,, . .	850	0·012
Plain ,, ,, ,, . .	55	0·0580
,, ,, ,, ,, . .	640	0·0148
Ten-face fluted rubber bearing, steel shaft	34·77	0·0253
,, ,, ,,	69·50	0·0158

According to these figures an increase in the bearing load results in a lower coefficient of friction, which is contrary to what is usually experienced with metal and other bearings. Busse and Denton (1932) state that frictional measurements carried out on metal bearings have shown that, over a limited range, a decrease in the viscosity of the

* General Rubber Goods Division, Dunlop Rubber Company, Ltd.

lubricant is accompanied by a decrease in the coefficient of friction, provided perfect film lubrication exists, and also that the results obtained have been supported by mathematical examination of the problem previously undertaken by Reynolds (1886) and also by Harrison (1922). For any given conditions, as the viscosity of the lubricant is reduced in a bearing, the thickness of the lubricating film will eventually become less than the height of any irregularities in the metal surfaces and metal-to-metal contact will result, thus increasing the friction in the bearing. Rubber, however, being flexible and deformable, can accommodate itself to such irregularities without destroying the continuity of the film and in this way the surfaces of the bearing can be kept separated by a film of lubricant which is less than that necessary for equivalent metal surfaces. It is this feature which enables water to be used as a lubricant in rubber bearings.

The action of a water-lubricated rubber bearing has been explained by Annis (1927) as follows: In any revolution of the shaft each single point will pass over one or more of the rubber faces which are separated by grooves. As the edge of each face is rounded, and owing to the resilience and deformability of the rubber, the load on each face varies from zero to a maximum value. Owing to this, the lubricating film of water is formed at a point of low pressure, and because of its low viscosity in comparison with oil, when a film is once formed it is dragged by its adhesion to the shaft through the length of the face, provided the speed of the shaft is in excess of a critical value depending on various conditions, but which can be safely claimed to be very low. For this action it will be obvious that no wiping edge should be presented to the shaft at any point by the rubber.

The advantage which rubber possesses in this way is considerably increased when the water used for lubrication contains grit or dirt. Owing to the ease with which the rubber is displaced and its resilience, the particle of grit does not lodge in the bearing but is depressed into the rubber without cutting and is then rolled by the rotation of the shaft into the adjacent groove, and so washed away. In addition, owing to this deformation the unit load on such particles approximates to that on the surrounding rubber, whereas on a hard surfaced bearing the load can rise to a considerable value before a particle is forced into the bearing or shaft surface. It therefore follows that any scoring action in a rubber bearing is considerably less than that possible for a hard surfaced bearing, and that the shaft remains highly polished instead of becoming deeply scored and perhaps grooved.

This advantage of a rubber bearing is not secured by the sacrifice of life. Rubbers having the necessary softness and resilience are known which possess extraordinary resistance to the combined cutting action

of abrasives, as is shown by their use for suction and dredging hose, sand-blast hose, the lining of chutes and launders where, under severe conditions, a life considerably in excess of that of special steels has been obtained. Two examples may be quoted to illustrate the remarkable resistance to the action of water containing sand and grit. In one case, rubber bearings fitted in a centrifugal sand pump were reported to be in perfect condition after the pump had handled 10,450 tons of coarse sand in 693 hours (1924). Previously, when using Babbitt and bronze bearings, only 125 hours' running had been obtained. In another case (Cutler 1927), bearings installed in a hydraulic turbine utilizing sand- and silt-laden water gave a much greater life than lignum vitæ bearings, which wore away in a comparatively short time. It would appear that if the grit is of the water-borne type, such as river silt or sea sand, there is little or no effect on the wear of the bearing.

According to Annis (1927) the development of the rubber bearing has resulted largely from the advent of the screw propeller in ships, where a bearing was required capable of satisfactorily carrying the load of the propeller shaft under the severe conditions existing at the stern tube or strut. Many designs had been evolved to exclude water and foreign matter from oil-lubricated bearings, which, until the advent of rubber bearings, were superseded by lignum vitæ, as this was found to be the most suitable material then available. Although ensuring a trouble-free bearing for a time, there was always the possibility of a scored journal or shaft liner with consequent excessive wear and this risk was considerably reduced when rubber bearings were introduced.

As illustrating the service obtained by rubber bearings in comparison with lignum-vitæ and other types, no sign of wear on either propeller shaft or bearing was reported (Schaphorst 1927) on a Diesel-propelled tug after 10 months' service. In another case (Annis 1927) the wear on the bearings carrying a 10-inch shaft was found to be only $\frac{1}{16}$ inch after a run of 35,000 miles, and no significant wear on the shaft was noted. Other observers (Anon. 1936) have reported that a rubber bearing gives a relative life of from four to ten times that of other types, including lignum vitæ.

Rubber bearings have also been extensively used for borehole centrifugal pumps where the pump is suspended by its own pipe work at the bottom of the bore hole and driven by an electric motor at the surface through a shaft running down the centre of the rising main. The shaft is supported at intervals by bearings, and rubber bearings, generally lubricated by the water being pumped, have been found a most satisfactory type for these conditions. In such pumping systems the loss of power caused by friction on the bearings is extremely small. Little trouble is experienced through rusting or corrosion of the shaft

if the rubber bearing is in continuous use and shafts made of "rustless" steels or even drawn nickel have been used. It is a common practice to fit a sleeve made of a non-corrosive metal on the shaft at the position of the bearing.

There appears to be no upper limit to the speed at which a rubber bearing can be used. The highest is probably that attained by *Miss England II* when a peripheral shaft speed of 4,330 ft. per min., corresponding to over 12,000 r.p.m., was developed (Schaphorst 1927). There is, however, a lower limit of speed which will depend on the load on the bearing. Shaft speeds as low as 100 ft. per min. with a load of 50 lb. per sq. in. have been quoted. At lower speeds, however, forced lubrication is recommended unless the load on the bearing is very low. Marine bearings for propeller shafts are seldom run at pressures on the projected area greater than 30 lb. per sq. in.

Contrary to what one would at first expect, the deflexion of a shaft in a rubber bearing under a safe load is comparatively small. Haushalter and Moffitt (1935) have quoted examples illustrating this. The shaft of a hydraulic turbine in a rubber bearing 9 inches in diameter, 30 inches long, for example, only deflected $\frac{1}{64}$ inch under a load of 1,000 lb. The oscillation of such shafts was stated to be within 0·006 inch when the bearings are new and it was considered that they would run for years before the wear in the rubber bearing allowed an oscillation of 0·012 inch, even when there was abrasive material in the water. The shaft deflexion in a rubber bearing in a ship, due to the shaft weight and the weight of the propeller, was found to be of the order of 0·028 inch, the normal working bearing pressure being 25–30 lb. per sq. in. of the projected area. This deflexion is of no moment because the clearance between shaft and bearing is increased only very slowly owing to the slow wear of the rubber.

Rubber bearings have operated successfully under loads of 600–800 lb. per sq. in., but for such high loads it is advisable that the full load be not applied until after the shaft has attained a peripheral speed of about 500 r.p.m., and sufficient water must flow through the bearing so that the temperature of the lubricating film is kept sufficiently low.

The length of a rubber bearing depends upon the load condition when it is in use. For centrifugal pump bearings, where there is practically no load on the bearing, a length equivalent to $2\frac{1}{2}$ to 3 times the diameter is sufficient. Marine bearings for propeller shafts have been made five times as long as the diameter.

Types of Rubber Bearing. For a rubber bearing the rubber is used as a lining in a metal tube or sleeve, which is usually of brass or bronze and may or may not be split. Special methods are used to

ensure that the rubber is satisfactorily bonded to the metal and adhesions as high as 600 lb. per sq. in. have been quoted. Generally it will be found that the rubber will itself tear before the bond gives way. The inside surface of the rubber is grooved to a much greater extent than the oil grooves in a metal bearing, to allow the unrestricted passage of water for lubricating and cooling.

Several types of rubber bearing have been developed, the main differences being found in the design of the rubber lining with regard to its effect on the passage of the lubricating water. In some cases

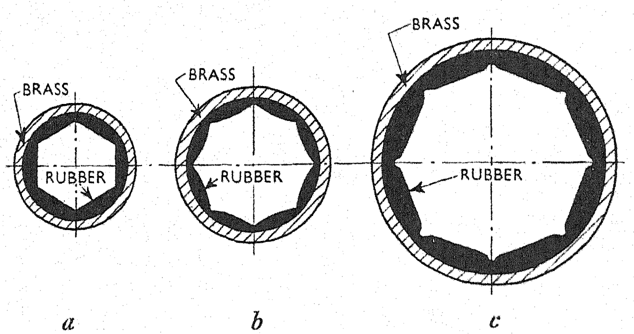


Fig. 1. Types of Rubber Bearing

these designs have been the subject of patents, and the following briefly summarizes the main types of interest:—

(1) A rubber bearing with aperture of hexagonal cross-section. This type has had several modifications (Fig. 1).
(2) One or more spiral grooves in the rubber lining (Fig. 2).
(3) A large type rubber bearing made in sections and containing helical lubricating grooves and annular slots for inlets and outlets of water (Fig. 3).
(4) A bearing consisting of a metallic sleeve having on its inner surface a number of longitudinal bearing members, individually removable, each consisting of a rigid portion for attachment to the sleeve and a resilient rubber portion to contact with the shaft. By slightly altering the width of the strip, bearings of different sizes can be lined.

Service conditions will determine the type of bearing to be used. For general industrial use the one illustrated in Fig. 1 *b* and *c*, having longitudinal grooves, is most suitable. For bearings where the lubricating water must be forced through, as in some propeller shaft bearings, helical grooves would be used. For comparatively heavy loads helical grooves placed close together are preferable, so that the

length of rubber surface in contact with the shaft at any one point is as short as possible. As compared with the longitudinally grooved type, this design offers a greater area of contact for supporting the load.

One of the main advantages of the rubber bearing is that, on account

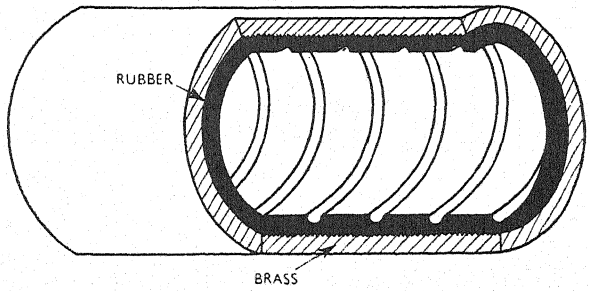


Fig. 2. Rubber Bearing with Spiral Grooves

of the resilience of the rubber wall, a shaft will operate without pounding, thus considerably reducing vibration, noise, and the effect of misalignment. This is of particular advantage for high-speed shafts, and a suitably designed bearing, combined with the use of rubber of suitable type, may permit a shaft to turn on a centre of gyration which

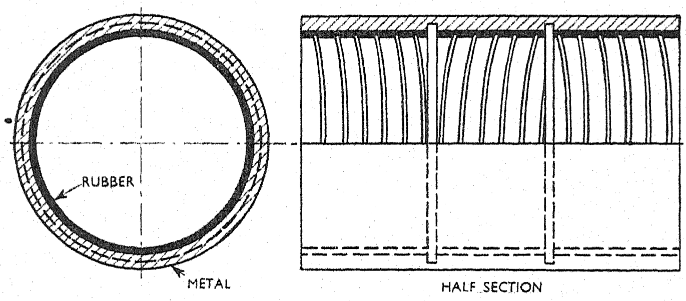


Fig. 3. Rubber Bearing with Helical Grooves and Annular Slots

may not coincide with the geometric centre, thus reducing the magnitude of the dynamic stresses set up by the revolving shaft. For this reason no useful purpose is served by placing a rubber bearing alongside or in conjunction with external rigid bearings, particularly if the object is to damp out vibration.

The degree of cushioning required often determines the thickness of the rubber lining used in the bearing, and the relation of these two factors is a subject on which a considerable amount of research remains

to be done. Much depends on the speed, diameter, and balance of the shaft and the arrangement of other bearings in the transmission. Since the wear on the rubber is usually extremely small there is no need for thickness solely from the point of view of wear. It is probable that in certain cases the functioning of a rubber bearing could be considerably improved by the use of a thinner lining, thereby reducing the amount of whip which could take place in a shaft.

It is obvious that the selection of the type of rubber to be used in such bearings requires careful consideration. Its hardness will depend on the general conditions of use, particularly the bearing load. A very hard rubber is unsuitable as, in addition to its being readily cut away by particles of sand or grit, the coefficient of friction obtained more nearly corresponds to those obtained with metal bearings. The rubber used must be resilient, tough, resistant to cutting and wet abrasion, whilst its permanent set must be as low as possible. In addition it should be of such a nature that, if necessary, it can be ground accurately to size with a suitable fine surface.

The requirements for the successful operation of rubber bearings are briefly as follows:—

(1) The size of the bore must be accurate and must give sufficient clearance for an easy running fit.
(2) Owing to the low heat conductivity of rubber, any frictional heat must be carried away by the lubricating water. A continual flow of cooling water is therefore imperative.
(3) The shaft and the bearing surfaces must be smooth and unscored.
(4) The lubricating grooves must be placed sufficiently closely, so that the temperature of the water film is not raised excessively.
(5) The bearing must never be run dry, even at starting.
(6) Oil and grease, especially mineral oil, even in small quantities, can often be detrimental to rubber bearings as they attack rubber and cause "tackiness." It is possible, however, by suitable compounding, to give the rubber considerable resistance to oil, and important developments are taking place in this respect.

Some of the uses to which rubber bearings can be applied have already been described, but their application covers a very wide field. They have given satisfactory service over considerable periods of use in under-water marine work, high-speed motor boats, hydraulic turbines, centrifugal pumps, agitators, washing machines, and domestic and industrial liquid-handling equipment. For pumping systems dealing with drinking water and for many solutions or fluids used in

the preparation of foodstuffs or beverages, rubber forms an ideal bearing material.

References

ANNIS. 1927 *Marine Age*, May, quoted in 1927 *India-Rubber Jl.*, vol. 73, p. 1044.

ANON. 1936 *India-Rubber Jl.*, vol. 91, p. 662; 1936 *Rubber Age* (London), vol. 17, p. 152.

BUSSE and DENTON. 1932 *India-Rubber Jl.*, vol. 84, p. 347, "Water-lubricated Soft-rubber Bearings".

CUTLER. 1927 *Rubber Age* (New York), vol. 20, p. 555.

HARRISON. 1922 Glazebrook's Dictionary of Applied Physics, vol. 1, p. 376.

HAUSHALTER and MOFFITT. 1935 *The Machinist* (London), vol. 79, p. 342 E, "Water-lubricated Rubber Bearings".

REYNOLDS, O. 1886 Phil. Trans. Roy. Soc., part 1, p. 194.

SCHAPHORST. 1927 *India-Rubber Jl.*, vol. 73, p. 592.

CONTRIBUTION TO THE THEORY AND PRACTICE OF BEARINGS

By Henri Brillié*

With modern knowledge it should be possible to determine in advance the conditions under which bearings work. Some of the problems presented by bearings, however, are not susceptible of mathematical treatment; while experience in itself is insufficient, as the results depend on too many factors, the influence of which varies according to working conditions. The problem is very complex for a number of reasons. Amongst these may be mentioned: the necessity for making various assumptions when integrating; the possibility that the film can be dispersed at any moment under the effect of the upstream pressure; the limited width of the actual film and the existence of lateral regions where pressure cannot develop to the same extent as in the central region. Further, important data relating to partial films are lacking. The effect of temperature on viscosity has also to be considered, and while it is supposed that the motion of the lubricant is streamline, certain experiments indicate that turbulent areas are present. The effect of oiliness still requires mathematical expression, so that it can be introduced into the calculations. In addition, the phenomena of oiliness are interlinked with phenomena resulting from irregularities of the bearing surfaces, and these, too, still have to be taken into account mathematically. These various factors will be dealt with separately.

Fluid Friction. The phenomena of fluid friction are due to the reciprocal action of the molecules. The matter in the molecule is negligible, and the molecule itself can only be considered as a field of force (Woog). In studying the film it is assumed (streamline state) that the molecules are assembled in superimposed layers which can slip easily one over the other. The cohesion of the liquid results from the fields of force which constitute the molecules. The molecules possess a vibratory movement, the energy of which corresponds to the temperature. The average distance between the centres of the fields of force is constant for a given temperature but increases rapidly with the temperature. Decrease in the viscosity of liquids with increasing temperature can thus be understood.

The fundamental equation of the theory of viscosity (Reynolds)

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depends on the supposition that one element of a layer moves in parallel to a mobile bearing surface, and that the resultant of the mutual reactions of these layers a, and of the upstream and downstream pressures $-b$, is equal to the inertia c, i.e. $a-b=c$. Let $c=0$ and $a=b$, that is to say, the film is being formed or if it is already formed, it continues to exist. Another possibility is that $b=0$ and $a=c$, a case which applies to a film which, when exposed to pressure, ceases to form or to exist, i.e. it is dispersed. This dispersion can happen at any moment. If it is assumed that this does not happen, and that b does not equal 0, the argument can only be continued by neglecting the value of c, i.e. by neglecting all hydrodynamic effects, in other words, all effects other than the purely kinetic effects, such as the hydrodynamic effects resulting from the wedge shape of the film or from a change in direction of the fluid particles. This leaves the treatment in the state of a first approximation which could only be properly justified by experiment. It has, however, been shown (Brillié 1929) that, when the film has actually formed, the hypothesis that $c=0$ is a sufficient approximation for many practical cases.

In order to integrate the equation $a=b$, it is assumed that, throughout the height of any transverse cross-section, the pressure is the same in all the layers. This hypothesis is probably true for a film some hundredths of a millimetre thick. However, this is merely apparent, and the hypothesis should be examined on the same scale as the element under consideration, i.e. on the scale of the molecules. The film could thus be represented as a superimposition of individual grains between which there exists a certain cohesion. These grains lie on a moving surface and the layer is limited by the fixed surface. If, for example, the grains were considered to have a size of the order of a centimetre, i.e. they are enlarged 10^7 times, then, as the film is, say, 0·01 mm. thick, it would appear to be a layer of grains 100 metres high. A bearing 5 cm. wide would thus correspond to a moving surface 500 kilometres wide. Can it be said that, during the movement of this moving surface, the pressure is the same at any point of a cross-section which is 100 metres high ? There is no reason *a priori* to admit such a hypothesis for the 10,000 layers which constitute the layer under consideration. This equality of pressure, however, will be assumed, so as to avoid mathematical difficulties, but it should be remembered that this constitutes a first approximation which may or may not correspond to reality. Experiment alone can decide whether the hypothesis is partially or completely justified.

Having assumed that $a=b$ and that the pressure is the same at any point in the same transverse section, the conclusion is that the speed curve is a parabola. The author has previously published (1929)

characteristic curves which enable difficult analytical integrations to be replaced by easy arithmetical summations. This will lead, without any mathematical difficulty, to a rigorous solution of the problem of the film, due reserve being made with regard to the hypotheses enunciated above.

Turbulent and Streamline Flow. In a film where streamline flow takes place in the direction of movement of the moving surface, the number of layers in a transverse section will depend on the height of the cross-section. In any film whose height varies there will be variations in the number of layers, so that there are interpenetration effects between the different layers. This interpenetration may take place uniformly, with a fairly regular distribution of all the elements of certain layers which disappear between other layers which remain. In this case, the hypothesis that there is a uniform variation of pressure in all the layers can be accepted. The interpenetration, however, may be irregular: certain layers can continue to exist with their number of molecules unchanged, so that they are unaffected by interpenetration, whereas, other layers, instead of losing their molecules to adjacent layers, give rise to turbulence. Here it is doubtful whether the hypothesis of the uniform variation of pressure in any point of a cross-section can be retained. It would not be surprising if experiment showed that, in certain areas of the film, there is a complete disturbance of the development of the pressures compared with the conditions resulting from these hypotheses. The surprise would be less if we considered the areas of return currents, whose reality will be seen from the speed parabolas (Brillié 1929a). This observation, derived from a kinematic study, is in formal contradiction with the fundamental hypothesis. The fluid, changing from direct to return current, passes through intermediate situations in which its movement becomes oblique and even perpendicular in relation to the direction of movement of the mobile surfaces. It may be thought that the half-circuit "direct current–return current" will be completed by another circuit "return current–direct current" which will give a general "oil roller" or swirling movement (Brillié 1929b). There is, however, an important difference, since, owing to the limited height of the cross-sections, there are considerable passive resistances, which become negligible with increasing height (generally 1 mm. or more) of the oil pockets ("bassins-relais").

The consideration of a layer which is theoretically stationary and whose fields of force intersect the fields of force of opposite direction belonging to adjacent layers of direct current and return current, together with the previous considerations, indicates the probability of

turbulent movements which are contrary to the hypothesis. These turbulent movements may have considerable influence on the development of pressure. This can be shown only by experiment, as has been done by Clayton and Jakeman (1936). In spite of these divergences between theory and practice, these causes of deviation will be neglected in the present connexion.

Constant Value of the Coefficient of Viscosity. At the same time as the development of pressures and passive resistances which correspond to the phenomena of viscosity, the vibratory energy of the molecules of lubricant, i.e. the temperature of the molecules, also increases in absolute correlation with the energy absorbed by the passive resistances. When the value of the coefficient of viscosity, the relative speed of adjacent layers and the speed of flow of a particular layer are known for one point, the rise in temperature of the molecules at that point can be calculated, assuming that the corresponding energy is not transmitted instantaneously to the neighbouring molecules.

Influence of the Limited Width of the Film. Considered in transverse section, there is necessarily no pressure in the film at the edges of the bearing. If it is assumed that the film is of infinite width, the curves of equal pressure are perpendicular to the direction of movement. For a brass of infinite breadth, the curves of equal pressure would appear as in Fig. 1 for the whole area where the pressures increase (upstream).

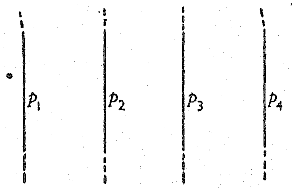

Fig. 1. Curves of Equal Pressure

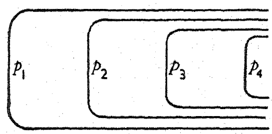

Fig. 2. Lines of Equal Pressure

The convex portion of the curve faces upwards and the variations in pressure can be determined (Reynolds's formula) by studying the variations in convexity of this curve in relation to the variations in height, constant flow and constant width of film being assumed. In fact, for a brass of finite width, there is, on each edge, a lateral band where the currents have the same pressure going from upstream to downstream, and this band assumes more and more significance. There is a tendency for the formation of lines of equal pressure (Fig. 2). For all the lines of current for which the pressure is constant from upstream to downstream, the speed curve is a straight line and the flow

is greater than in the case represented in Fig. 1. This greater flow in certain areas (two lateral bands) corresponds, for a constant average flow, to a continuous diminution of the width of the film. Actually, the lines of flow will be as in Fig. 3, and not as in Fig. 2. This general shape of the film can be seen on thrust collars with oil pockets, as the oil film forms in time an adherent varnish on the metal. In the case of Fig. 3, if the oil pockets were insufficiently close together the film

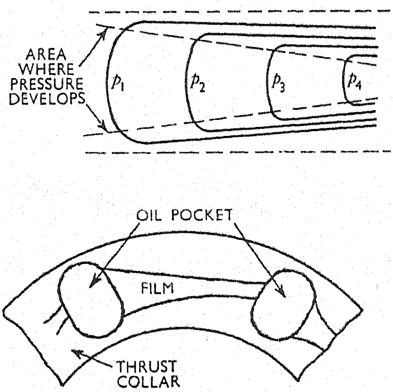

Fig. 3. Thrust Collar with Oil Pockets

would disperse owing to the increase in width of the lateral bands where there is no change in pressure. In fact, the decrease in breadth of the film takes place despite the development of pressures which should either cause the film to widen or destroy it. The contrary effect occurs owing to cohesion, but there is a limit to cohesion which would be exceeded were the oil pockets too far apart. This tendency towards the progressive decrease in width of the film would actually result, for a given thickness in film, in pressures which would be lower than the theoretical values, i.e. it would result in a lower thickness of film for a given load.

Oil Losses. Oil losses can occur at the edges of the film, thus reducing the flow and consequently the extent of the film. This could be taken into consideration by means of a suitable correcting factor.

Film Stability. Tendencies to instability of the film could be overcome by arrangements such as the wedge shape of the section, and the provision of grooves placed sufficiently close together and perpendicular to the direction of movement.

Conditions for Calculation. For a first approximation in calculations relating to the theory of viscosity, it is assumed that: (*a*) hydrodynamic phenomena can be neglected; (*b*) pressures are uniform in any point of the same transverse section; (*c*) the speed curve is parabolic; (*d*) the different layers are independent and the streamline state is continuous; (*e*) the coefficient of viscosity is constant; (*f*) the influence of the limited width of the film can be neglected; (*g*) loss of oil can be neglected, a constant flow being assumed; and (*h*) the film is stable. From considerations set out above the following general principles can be deduced.

General Principles. The film may not form or, if formed, may be dispersed by the pressures developed upstream. However, the film will form and exist in one area if that area is preceded upstream by a suitable nozzle ("adjutage") which ensures priming and if the area is established under conditions which prevent dispersion. If these two conditions are satisfied so that the formation and stability of the film are secure in a certain area, then, for a given flow, there will be a pressure variation at each point which will depend on the speed of the moving surface and the height (thickness) of the film at that point (Reynolds's formula). For a given flow and speed of the mobile surface, there will be a film thickness which will give the maximum change in pressure (Reynolds's formula). The best film in a given area will be that which has that particular thickness at each point, and thus will lie between parallel surfaces. From upstream downwards the film tends to narrow under the constant change of the extent of the two lateral zones which, on the edges, show no change in pressure. The result is that the film can disappear rapidly if special devices are not adopted. Experiment, in agreement with theory, shows that transverse grooves of suitable section and position prevent the dispersal of the film, or, what is the same thing, cause the formation of "elementary" or "partial" films which can benefit in the upstream area, as a result of the transverse groove, from the pressure gained downstream by the preceding film. Although pressure is not developed in the area round the groove, and although the grooves may cause local decreases in pressure, they are often advantageous as they ensure continuity of the film, which would otherwise cease to exist.

The Elementary Theory of Viscosity. With the reservations indicated above, the oil film can form between the bearing surfaces of the bushes in conformity with the conditions resulting from the theory of viscosity. If we continue to neglect the influence of variations of temperature, surface irregularities, and oiliness, simple formulæ can

be deduced which are sufficiently accurate to permit difficult analytical integrations to be replaced by simple arithmetical summations. In this way, a study can be made of films between parallel surfaces and in bearings, complete pressure films, partial films, the oil wedge, and so on—subjects which have been dealt with in previous papers by the author. Here, only the conclusions arrived at will be given.

The Hypothesis of the Speed Parabola. If it is assumed that the speed curve is a parabola and that the flow and width of the film are constant, the exact shape of the speed curve can be determined from the well-known properties of the parabola. Let H' be the height of the section separating the region of return currents (upstream nozzle) and the film properly so called, then the vertex of the parabola is on the fixed surface. Let h' be the height of the section for which the parabola is a straight line (maximum pressure). Let h'' be the height of the section where the film produces a negative restraint on the moving surface; then the apex of the parabola is on the mobile surface. This section marks the boundary between the film properly so called and what the author calls the "downstream adjutage." These heights of sections are related as follows:—

$$H'=(3/2)h'=2h''$$

Characteristic Curves. The hypothesis of the parabolic form of the speed curve leads, when the apex of the parabola is on the moving surface and the height of the section is H', to the two following relations:—

The variation in pressure in the direction of movement for all points of the section

$$=dp/dx=2Z(V/H'^2).$$

The passive resistances per square centimetre for the section

$$=f=2(Z)(V/H')$$

and, for the same flow and for any section of height h, to the two relations

$$dp/dx=2Z(V/H'^2)B \text{ and } f=2Z(V/H')A,$$

B and A being functions of the ratio h/H' which can be plotted once for all. These characteristic curves serve as a basis for the mathematical study developed in various papers by the author (Brillié 1929–37).

Films of Uniform Thickness between Parallel Surfaces. The constant thickness of the film being h, the speed parabola and the flow of the

film are mathematically defined by the relation h/H', as well as the corresponding values of the functions B and A, and we have (in practical units):—

The total pressure, equal to the load,

$$P=100\times Z\times(V/H'^2)\times a^2\times l\times B$$

The total passive resistance

$$F=0{\cdot}200\times Z\times(V/H')\times a\times l\times A$$

Elementary Films of Slight Eccentricity. In the case of the "elementary film" of a bearing, that is to say, of a film of limited length, for which θ, the angular position of the line of action of the load with regard to the line of centres, can be neglected, the film will be of fairly uniform thickness if there is almost no eccentricity, and, using the notation: R=radius of the bearing, p_s=pressure in kilogrammes per square centimetre of projected area $=P/2lR$, $u_I=Z(N/p_s)$, the preceding formulæ become

$$H'/a=0{\cdot}23\sqrt{u_I}\sqrt{B}$$

$$f=10^{-3}\times0{\cdot}46\sqrt{u_I}\times(A/\sqrt{B}$$

If the thickness n of the film is equal to H',

$$A=B=I$$

and the formulæ become

$$n'/a=0{\cdot}23\sqrt{u_I}$$

$$f=10^{-3}\times0{\cdot}46\sqrt{u_I}$$

The coefficient 0·23 will be a maximum and the coefficient 0·46 a minimum for partial films.

Elementary Films of Any Cross-Section. If the cross-section is not of uniform thickness, the film is divided into a number of elements for which the unit pressures, the total pressures, and the total passive resistances can be calculated. The only change in the above values for the total pressure P and the total braking effect F will be the replacement of the coefficients B and A by the coefficients resulting from the corresponding integrations (Brillié 1935).

Partial Films. In the case of partial films, the angular position θ of the line of action of the loads with regard to the line of centres is taken into consideration. To each element of the film is assigned the corresponding values of cos θ and sin θ, and the values of the com-

ponents of the total load are determined according to both the line of centres and the perpendicular direction. For the study of partial films, as well as of elementary films, further clarification is necessary. In fact, for these films, neither the upstream section nor the downstream section, nor the form of the speed curve for any section is known. To define the problem, it is assumed that the film ends at the portion of minimum thickness (line of centres), n, and three distinct hypotheses are made, namely, that $n=H'$ (hypothesis A), $n=h'$ (hypothesis B), and $n=h''$ (hypothesis C). Further, these three hypotheses may correspond to practical cases: hypotheses A and C are limiting hypotheses, and hypothesis B represents an average. Every practical case will interpolate between hypotheses A and C. The calculations and curves corresponding to these three hypotheses have been dealt with in a recent paper (Brillié 1937).

Nozzles. The priming of the film is produced by the development of pressures in the oil layer, before there is separation of the two bearing surfaces, owing to the phenomena of viscosity due to the return currents. If it is supposed that these phenomena occur up to a limiting cross-section of height h_c, the pressure per square centimetre at the downstream end, as well as the total pressures perpendicular to the direction of movement can be calculated for various sections. These pressures produce the priming of the film. The author has shown (Brillié 1935) that, taking the height of the limiting section as the relatively large value of 0·01 mm., and assuming nozzles 1 cm. long, a straight, wedge-shaped section, 1 mm. high at the entry, is almost without effect on the pressure perpendicular to the direction of movement; its effect on the pressure per square centimetre at the downstream end is slight; it is 10 times less than if the height at the entry was 0·1 mm. A parabolic section, or a circular section given by a "nozzle radius," is much more efficient than a straight section; for a height at entry of 1 mm., the pressure per square centimetre at the downstream end is three times greater. Further, the "over-clearance" of the bearing gives a very efficient starting effect; for a radius of 10 cm., an over-clearance of 0·1 mm. on the radius gives, under the above conditions (length of nozzle 1 cm.), a pressure of 900 kg. per sq. cm. at the downstream end, and a total pressure perpendicular to the surfaces of 100 kg.

Oil Wedge. This has been dealt with elsewhere by the author (Brillié 1928, 1929).

The Complete Film. The problem of the complete film is entirely defined mathematically by the fact that, starting from any section of

the film and coming back to the same portion after running over 360 deg., the pressure will be the same at the start and at the end. The results of the author's investigations have been expressed (Brillié 1929) as curves having as abscissæ the values of the coefficient of speed u which synthesizes the running conditions. The value of this coefficient is

$$u=(\mathrm{N}/p_s)\times(\mathrm{Z}/m^2)$$

where m is the clearance coefficient, equal to r/R, r being the clearance in hundredths of a millimetre, and R the radius in centimetres.

Effects of Temperature, Oiliness, and Surface Irregularities. These related phenomena are discussed in the author's paper in Group IV of this Discussion.*

Theoretical and Experimental Investigations. Among the numerous experiments which have given results comparable with the indications given by the author's theoretical curves and formulæ, special mention may be made of the work of Kingsbury (Hodgkinson 1929), Clayton and Jakeman (1936), and Thomson (1936).

Kingsbury's Results. Kingsbury worked with air. The coefficient of viscosity of air is independent of pressure and very nearly independent of temperature; under these conditions, experience is more likely to agree with theory, which usually neglects the variations of the coefficient of viscosity with increase in temperature of the film. The measurements, made with high accuracy and for high journal speeds, disclosed the pressures existing at various points of the liquid layer. The curves obtained by Kingsbury agree with the author's theoretical curves relating to the complete film.

Clayton and Jakeman's Results. Comparison of the experimental results obtained by Clayton and Jakeman (1936) for complete clearance bearings with the author's results leads to the following conclusions:—

Coefficients of Friction. All the experimental determinations of coefficients of friction give results which, within the limits of experimental accuracy, correspond to the theoretical curve relating to the complete film.

Relative Film Thickness. In all the experiments regarding the relative thickness of the film, the results agree with the theoretical curve corresponding to the partial 60 deg. film (hypothesis B).

Attitude. Consideration of the graph giving the attitude for various values of relative thickness of the film suggests that the partial films

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* See vol. 2.

corresponding to the National Physical Laboratory experiments are, for the development of pressure, of about 105 deg., the running conditions being intermediate between those of hypotheses B and C: the values of the coefficients for the relative thicknesses of the film are 0·07 and 0·05. However, agreement between theory and experiment for the group corresponding to the smallest clearance (r=0·019 mm., m=0·75) can be arrived at by supposing that the film, having an arc of about 105 deg., is formed 125 deg. in advance of the section where the thickness is a minimum, and is dispersed about 20 deg. upstream from that section.

Under the conditions in question, it appears that the experimental curves agree with the theoretical curves for the coefficients of friction, the relative film thicknesses, and the attitude.

Thomson's Results. Comparison of the results obtained by Thomson and by the author leads to the following conclusions:—

As shown by Table 1, there is generally a considerable difference between the extent of the bearing surface as calculated, and that actually obtained.

TABLE 1. ACTUAL AND CALCULATED BEARING SURFACES

Calculated bearing surface a_s, deg.	Width of bearing, cm. (D=6·3) l	Coefficient of speed u	Actual film a', deg.
180	10·1	56	165
180	10·1	18	160
180	10·1	4·7	135
90	10·1	18	90
90	10·1	4·7	83
60	10·1	—	55
90	5·1	9·4	83
90	2·5	9·4	85

Under the experimental conditions, the section of maximum pressure h' corresponds very closely to the direction of the load; it occurs slightly downstream from that direction by a maximum of 5 deg.

The section of minimum thickness sometimes becomes fictitious (a_s=180 deg., l=10·1 cm., u=65·4), that is to say, the angle (180 deg. $-\phi$) is greater than half the calculated bearing surface.

The section of minimum thickness may fall in region D (Fig. 4),

that is to say, downstream from section h''. This is the case for small coefficients of speed, or arcs exceeding 90 deg. In this case, the value of the ratio h'/n is greater than 1·3 (about 2 in the experiment).

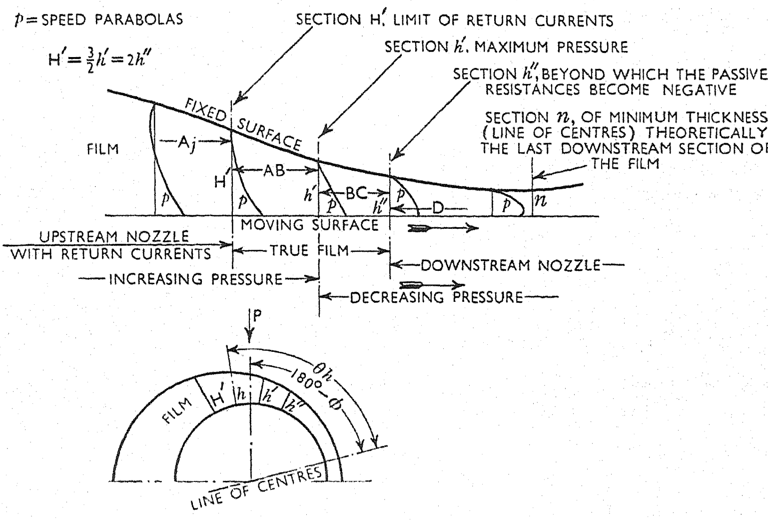

Fig. 4

P, load.

$$h/n \quad = 1-(1-n/r)\cos\theta_h/(n/r)$$
$$\cos\theta_h = (1-h/r)/(1-n/r)$$

Coefficient of speed $\quad u = (\mathrm{N}/p_s)(\mathrm{Z}/m^2)$

Theoretical coefficient of friction $\quad f_t = 10^{-3}\times \mathrm{K}_f m\sqrt{u}$

Theoretical relative thickness $\quad n/r = \mathrm{K}_{n/r}\times a/\mathrm{R}\times\sqrt{u}$

In Thomson's experiments, n is either in the area BC or in the area D.

Results of the integrations:—

	Values of K_f and $\mathrm{K}_{n/r}$ for $20 < u < 100$			
	Optimum film of uniform thickness	Partial films		
		Hypothesis A	Hypothesis B	Hypothesis C
K_f . .	0·46	0·46	0·54–0·7	0·47–0·54
$\mathrm{K}_{n/r}$.	0·23	0·16	0·12–0·07	0·08–0·04

For greater values of the coefficient of speed, or for a smaller arc, the section of minimum thickness n is in the region BC, between sections h' and h''. The ratio h'/n is then less than 1·3.

For a definite bearing surface a_s, and a definite width of the bearing, the coefficient of relative thickness $K_{0 \cdot n/r}=(n/r)(I/\sqrt{u})(R/a_s)$ is practically constant for any value of the coefficient of speed.

The value of the coefficient of relative thickness, which should at most be 0·23 for the optimum film of constant thickness, and should be about 0·10 for hypothesis B, and about 0·04 for hypothesis C, varies from 0·048 to 0·054 for intermediate conditions between hypotheses B and C ($h'/n<1{\cdot}33$) and from 0·012 to 0·017 for extrapolation with regard to hypothesis C ($h'/n>1{\cdot}33$).

The coefficient $K_{0 \cdot n/r}$ actually varies in the way indicated by theory.

The angular positions of the line of centres with regard to the direction of the load are in satisfactory agreement with theory, taking into account the position of the portion of the film with minimum thickness with regard to the areas B, C, or D.

The considerations regarding the coefficients of friction should indicate, as with Clayton and Jakeman's experiments, regions of the film where passive resistances are developed without any corresponding development of pressure.

The actual development of the film should only be about 0·9 of the calculated development for 60 deg. and 90 deg. brasses, and about 0·8 for 180 deg. brasses.

The "experimental coefficients" * of friction are thus about 1·22 for 60 and 90 deg. brasses, and 1·66 for 180 deg. brasses (width of bearings, 10·1 cm.).

The influence of decreasing pressure on the edges is shown by the difference observed between the above figures and those corresponding to the recorded pressures.

Conclusions. To sum up, if all the factors of the theory of bearings (oiliness, surface irregularities, variations in temperature of the film, phenomena of turbulence, the influence of the edges), which have to be neglected in order to arrive at a simple theory, are considered, the agreement between theory and experiment will be found to be satisfactory. The theoretical formulæ, together with "experimental coefficients", permit of interpolation, and even extrapolation to a certain extent, with regard to experimental conditions. These interpolations and extrapolations could well be made much more extensively, were it realized that it is possible to apply the principles of mechanical similarity to the running conditions of bearings.

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* The author calls "experimental coefficients", the factors by which the theoretical coefficients must be multiplied in order to obtain the observed coefficients.

The results given by the experiments of Clayton and Jakeman and of Thomson clearly confirm the two following general principles:—

(1) The development of pressures and of passive resistances does not occur over the whole bearing surface according to the conditions resulting from the integration of Reynolds's formula. As this development (particularly of the pressure) only occurs on part of the surfaces, the result is the production in the bearings of a region without utility for running, having the sole effect of increasing the passive resistances. It would be evidently advantageous to avoid this harmful effect: a simple solution being to provide, in the appropriate region, an oil reservoir, which need be only 1–2 mm. deep.

(2) Thomson's results show clearly that there are difficulties in the way of obtaining films of large arc. Even with small bearing surfaces, such as a 60 deg. arc for bearings 6 cm. in diameter, the film does not extend over the corresponding length, limited to 3 cm. in all. The arc is only about 54 deg. instead of 60 deg. These experimental observations are in full agreement with theory.

When a large effective bearing surface is required, it would be advisable to provide for the formation of successive elementary or partial films not more than a few centimetres wide. This experimental result agrees with the author's observations, and with his practical experience which has led him to the use of "transverse oil-pockets". Practice has shown it to be advantageous to place these pockets so that the space between them is roughly equal to the width of the pocket; usually the width between the edges is from one to three times the width of the pocket. This apparent reduction of the bearing face is not disadvantageous if the oil supply is ample, the film being capable of supporting pressures much greater than 100 kg. per sq. cm., under the condition of limited length which is discussed above.

References

Brillié, H. 1928 Bulletin technique du Bureau Veritas.

1929a Association Technique Maritime.

1929b Bulletin technique du Bureau Veritas, "Étude cinématique du film d'huile".

1929c Trans. Inst. of N. Architects, vol. 71, p. 338.

1935 *Science et Industrie*, "La technique du graissage".

1937 Association Technique Maritime.

Clayton, D., and Jakeman, C. 1936 Proc. I.Mech.E., vol. 134, p. 437.

Hodgkinson, F. 1929 Proc. I.Mech.E., p. 843.

Thomson, A. S. T. 1936 Proc. I.Mech.E., vol. 133, p. 413.

Woog, P. 1926 "Contribution à l'étude du graissage. Onctuosité. Influences moléculaires" (Delagrave, Paris).

STEAM TURBINE LUBRICATING SYSTEMS AND FIRE RISK

By J. P. Chittenden, M.I.Mech.E.*

Forced lubrication was applied to steam turbines from the earliest days, and although the system was then confined to the duties of lubrication, developments in the operation of steam admission valves took place and eventually it became the standard practice to utilize a portion of the lubricating medium under pressure to operate servo-motors controlling these valves. Several variations of this system are used, but the underlying principles are the same, namely, to employ a higher pressure supply for relay operation than is required for lubrication, and so provide servo-motors of ample power. For the operation of relays the intermittent use of the gear and the necessity for instantaneous response demand that there shall always be available a sufficient supply of oil to meet these sudden requirements without an appreciable fall in pressure. For the lubricating system proper sufficient oil must be provided to carry away the heat at the journals. Heat taken up by the lubricating oil must be dissipated and so an efficient cooling system has to be provided. The importance of the effective disposal of this heat will be appreciated when it is realized that for a large modern turbo-generator the heat to be extracted from the oil is of the order of 2,000,000 B.Th.U. per hr. Further, the rate of circulation of the oil through the system should bear such a relation to the total capacity that ample opportunity is provided for the oil to be so treated that impurities, entrained gases, water, etc., are disposed of. Fig. 1 (*a* and *b*) indicates the approximate total oil capacity and pump capacity for units of various sizes, based on the general practice obtaining in this country.

Systems fulfilling the above requirements may be divided into two classes, one in which the lubricating and governing oil is supplied by a separate system which may serve more than one turbine, and the second in which the oil pump, cooling system, and other components form an integral part of the turbine installation. The former is usually employed in this country for marine turbo-electric installations only, and the application of this method to land use on a large scale has apparently only been made, some years ago, at the Southwick Power Station of the Brighton Corporation.

In the more usual system, where each turbo-generator has its own

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oil supply, the pump is mechanically driven direct from the turbine shaft. Common practice is to pump the total quantity of oil to the higher pressure required for operating the relays, i.e. about 40 to 80 lb.

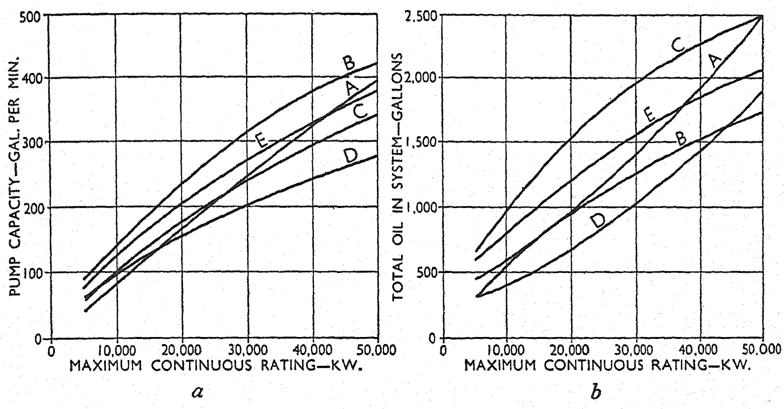

Fig. 1. Approximate Total Oil Capacity and Pump Capacity for Turbo-Generators of Different Sizes

per sq. in. This pressure is reduced to about 5 to 15 lb. per sq. in. for supply to the bearings.

In a typical lubricating system for a machine having a single pressure pump (Fig. 2), oil is drawn from the tank by the main oil pump, which

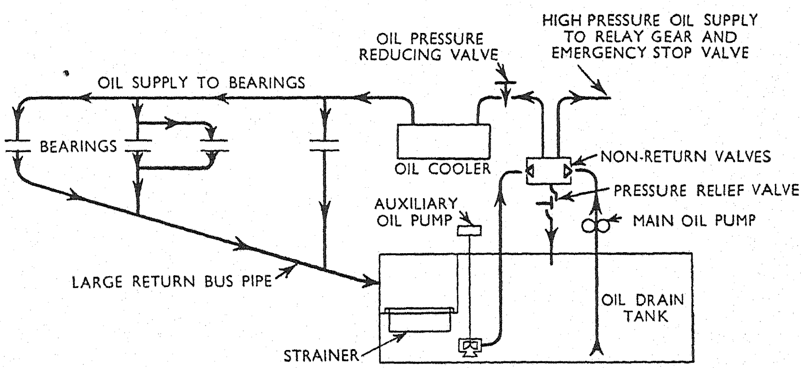

Fig. 2. Typical Lubricating System

is of the displacement tooth type. This pump may be submerged or may have a slight suction head. From the pump, oil is delivered into a distribution box through a non-return valve. In the event of a breakdown on the main pump, this valve prevents the oil supplied by

the auxiliary pump from flowing back through the disabled main pump. The intermittent use of the relay gear which is supplied from the distribution box calls for sudden demands on the high-pressure oil supply for governing, and so the distribution box is provided with a relief valve designed to pass this quantity of oil at times when the relay gear is not operating, and set to relieve at the pressure required for governor operation.

Oil for the lubricating system is passed from the distribution box through a pressure-reducing valve to the coolers, and thence to the various bearings. The reducing valve may be of the hand-controlled or automatic type, capable of the slight variation necessary to maintain the correct quantity of oil for the bearings when the oil is cold and viscosity high.

On the return side the oil leaving each pedestal is led into a large-bore return pipe, which must provide ample area to allow for the increased volume of the oil due to frothing and to assist in the dispersal of the entrained gases. From the return pipe the oil passes through strainers back into the tank. In the system illustrated the strainers are shown in the oil return line. This method enables provision to be made so that if the strainers become choked, the oil, after building up to a certain level in the strainer chamber, flows over a weir into the tank so that the circulation is not interrupted or even diminished.

An auxiliary oil pump is included to provide the necessary lubrication and pressure oil relay supply for starting and stopping. This pump generally has the same capacity as the main pump, and also acts as an emergency standby to the main pump. It may be steam- or electrically driven, and on large sets both alternatives have been employed as a double safeguard. Steam-driven auxiliary pumps are usually of the centrifugal type and deliver into the common distribution box through a non-return valve from which the oil follows the course previously described.

On some large machines separate high- and low-pressure pumps have been employed; the high-pressure pump then delivers into a chamber which relieves into the low-pressure system fed by its own pump. By this means the loading on the gearing driving the oil pumps is reduced, since only the oil required for relay operation is pumped to the higher pressure. In such a system the auxiliary pump for simplicity supplies both relay and lubricating oil at high pressure.

The arrangement and disposition of tanks is usually a matter of convenience to the general layout. The essential feature is that the capacity of the system must be sufficient to provide a reasonable settling time for the separation of gases, etc. One change of oil through the system every 6 to 8 minutes is generally allowed in large modern

machines. This involves oil capacities such as are shown in Fig. 1*a*. To accommodate such quantities in one tank at the steam end presents difficulties on the larger machines owing to limitations imposed by the foundation structure, and a second tank B (Fig. 3) is frequently used to augment the capacity of the pedestal tank A. This second tank may be totally enclosed and therefore can be placed in any convenient position below floor level.

The oil, having passed through the strainers into tank A, is led into the bottom of tank B and the suction pipe to the main oil pump is taken

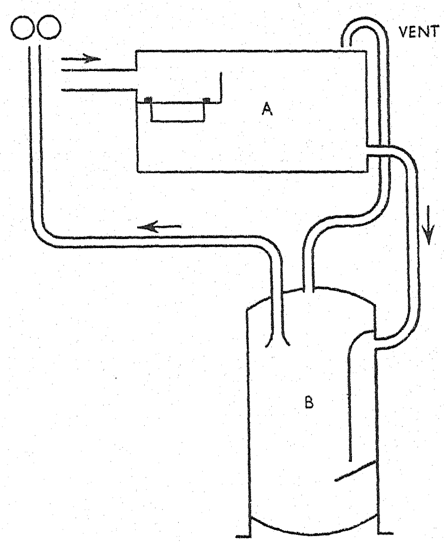

Fig. 3. Method of Increasing the Tank Capacity of the Lubricating System

from a point a little below the top of tank B in order to avoid drawing oil from the aerated region which exists in the upper portion of the tank.

With regard to the regulation of oil to individual bearings, it is the practice in some cases to control this by means of valves or throttle plates, but as such restrictions involve a risk of choking, the author prefers that the desired control be obtained by attention to the bearing clearances, and as the wear of correctly designed turbine bearings over many years is negligible, no further attention should be necessary.

The materials with which oil comes in contact are mainly cast iron or steel, but there are certain items in a turbo-generator where it is

necessary or advantageous to employ non-ferrous metals and in these cases care should be taken that they are not subject to oil at very high temperatures as such metals may act as catalysts under certain conditions. Further reference to this will be made when discussing fire risks.

Special care must be taken to remove all moulding sand, scale, etc., from the internal surfaces of iron castings used as oil passages or in contact with oil. Apart from possible damage that may be done to the working parts of the system, any foreign matter which may be deleterious to the oil must be removed. The same remarks, of course, apply to the oil piping and tanks. As a further precaution, some turbine manufacturers cover these surfaces with an oil-resisting paint, but on this point there appears to be a diversity of opinion. Special oils are now available for flushing out the oil system before putting in the final charge of oil, which is a practice to be commended as the

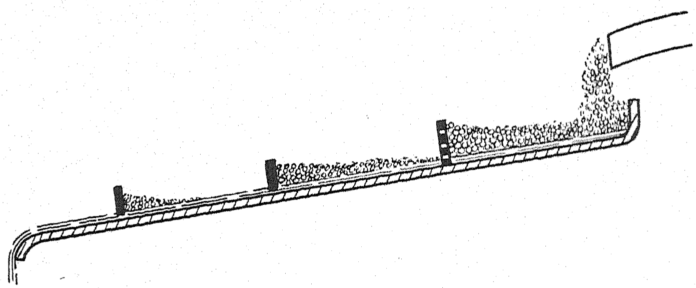

Fig. 4. Baffle System for the Removal of Gases from Lubricating Oil

washing of these surfaces with hot oil is a further precaution against the presence of foreign matter.

The removal of gaseous impurities in the oil is extremely important. These may comprise air, water vapour, and the chemically "cracked" lighter fractions of the oil which are acid in nature and promote the formation of sludge. Much of these vapours can be removed by suitable venting, the use of large return pipes, and efficient baffling in the tanks to facilitate the release of gases and the settling out of water and sludge.

A further method, which has been tried by the author, with considerable success, consists of running the more gaseous oil very slowly over shallow trays with baffles so arranged that the froth is held back and the clear oil passes underneath the baffle (Fig. 4).

The incorporation of oil-purifying apparatus in the system cannot be dealt with here, but in passing it can be said that on most modern

sets separators of the centrifugal type are fitted, which are generally designed to bypass through the purifier about 5 to 10 per cent of the total quantity of oil in circulation, the periods of running depending upon the condition of the oil.

Fire Risk. The potential danger will be appreciated when it is realized that the oil can ignite at a much lower temperature than its open flash point, and further, if it is sprayed in a finely divided form the oil burns fiercely. Investigation has shown that a large proportion of oil fires owe their origin to a fine spray of oil impinging on a hot surface such as a superheated-steam pipe, ignition being due to an exothermic reaction between the highly volatile fractions produced by cracking and the atmospheric oxygen, and the consequent increase in temperature. These leakages are often due to failures of pipes, joints, and small fittings in the high-pressure supply, and particular attention should therefore be paid to their design.

Solid-drawn steel pipes should be employed throughout and cast iron and non-ferrous materials should be avoided. Particular care should be exercised in the design of all joints, bearing in mind the peculiar penetrating power of oil. Pipe joints commonly regarded as satisfactory for the pressures concerned, cannot always be relied upon to maintain a tight joint against oil, and it is becoming general practice to use at least the next higher table of standard pipe flanges and make all fittings suitable for pressures very much greater than the maximum pressure to which the oil is pumped.

As a further precaution against the spreading of oil, wherever possible the oil pipes should run in suitable channels below the floor level, and exposed pipes should be provided with shields, or cased pipes should be employed. Finally, every possible precaution should be taken to ensure that high-pressure oil pipes are placed as far as possible from hot steam pipes, valves, etc., and where such an arrangement is not practicable, shielding should be provided. The shields or other protective devices should be such that in the event of a failure, oil cannot be sprayed on to an adjacent machine, as it is known that this has caused oil fires.

The author would suggest that the greatest protection against fire is obtained by concentrating on the design of all pipes, joints, and fittings, with suitable shielding and other protective devices, rather than in fire-fighting equipment. By this it does not necessarily follow that fire equipment should not be fitted, and in this direction several effective methods have been evolved.

There is a very large measure of agreement amongst manufacturers on the general requirements of forced lubrication systems and the

precautions to be taken against fire risk. A study of the curves in Fig. 1 appears to show a diversity of opinion on pump capacities, etc., but it should be explained that for simplicity these have been based on kilowatts output, and do not, therefore, take into consideration such factors as size, number of bearings, surplus oil allowed for governing, or oil temperature rise allowed and capacity of cooling system. Again, the ratio of total oil to oil circulated is, of course, a matter of opinion and of the experience of the designer with the particular system and class of oil adopted.

The author wishes to thank the British Thomson-Houston Company, the Metropolitan-Vickers Electrical Company, Messrs. C. A. Parsons and Company, Ltd., Messrs. Richardsons, Westgarth and Company, the Brush Electrical Engineering Company, and the General Electric Company, for assistance given, and also the English Electric Company, for permission to publish this paper.

THE EFFECT OF SEIZURE ON THE SHAPE OF THE CROWN OF A BUSH, AND THE INFLUENCE ON SUBSEQUENT RUNNING*

By D. Clayton, B.Sc.†

The work described in this report arose from that described in a recent paper (Clayton and Jakeman 1936 ‡). It had been found that running a bearing to incipient seizure led to a change of shape of the crown of the bush, and it was desired to investigate the conditions under which this change of shape occurred. The journal bearing machine was the same as that described,‡ except that arrangements had been made to take the load off the bush completely for stopping

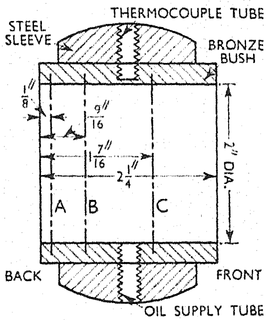

Fig. 1. Measuring Positions in the Bush

and starting, and for reversing the direction of motion of the machine when determining the value of the coefficient of friction. Also the oil was fed to the bush by gravity at a head of about 4 feet, the oil passing through a wash-leather filter into the feed tank. The apparatus used to measure the variations of diameter round the bush had a sensitivity of 0·0000018 inch per division of the reading microscope, and settings could be repeated to 1 division. The circumferential measurements were made at the three positions A, B, and C shown in Fig. 1. The bearing was heated so as to obtain a coefficient of friction–temperature curve. The friction decreases with increase of temperature due to decrease of viscosity, but reaches a minimum value and then

* Work performed for the Lubrication Research Committee of the Department of Scientific and Industrial Research.

† Scientific Officer, Engineering Department, National Physical Laboratory.

‡ Proc. I.Mech.E., 1936, vol. 134, p. 437.

begins to increase again. This rise of friction occurs at an increasing rate and the curve eventually becomes practically vertical; the machine then either stops or the driving belt is thrown off. At this stage there is an approximation to seizure of the bearing, and the temperature is referred to as the "seizing temperature".

Conditions of Test. The first experiments were made with a load of 180 lb. per sq. in. of projected area, at a speed of 500 r.p.m., and with a summer-grade motor-car oil, the diametral clearance being 0·006 inch. Seizure occurred at 200 deg. C. on the first occasion, and eventually was reached at 360 deg. C. The distortion of the bronze bush was very large with these high temperatures, however, and fresh tests were therefore undertaken. To keep the seizing temperature low the load was increased to 500 lb. per sq. in., the speed was reduced to 100 r.p.m., and the oil chosen was a thin (compounded) spindle oil with the following viscosity values: at 70 deg. F., 80·9 centistokes (327 sec. Redwood); at 200 deg. F., 5·8 centistokes (41 sec. Redwood). The diametral clearance was 0·0053 inch. The feed of oil was generally about 1 fluid oz. per min.

Run to Minimum Friction. As a preliminary to the main tests the bearing was run to minimum friction to see whether any contact occurred, as the explanation of the first rise of friction is of some interest. The test was actually carried to the stage at which the friction just began to rise, as shown for two runs at A and B in Fig. 2. The bearing was kept at the minimum friction for 2 and 7 hours respectively before cooling and then stopping. Owing to the low viscosity and speed there was little rise of temperature due to viscous friction, so heat had to be applied immediately; also the minimum friction was soon reached. The very low value of the coefficient of friction at the minimum is noteworthy.

Examination of the bush showed there to be very few scratches, but several brightly polished areas along the length of the bush, extending circumferentially from about 20 deg. on the outlet side of the crown in all cases, to about 5 deg. on the inlet side in the case of the largest area. Running in the opposite direction in a similar way (diagrams C and D in Fig. 2), the polished areas were rendered symmetrical about the centre line of the crown. On measuring the bush, there was a barely perceptible effect on the shape circumferentially.

It thus appears that high point contact is obtained when the friction begins to rise; further tests will, however, be made to provide more evidence. That the polished areas are eccentric with regard to the centre line of the crown is interesting as it indicates that contact is made at an attitude angle less than 180 deg.

Run to Seizure: Anticlockwise. The bearing was next run to seizure in the anticlockwise direction. The friction-temperature curve (curve *a*, Fig. 2) shows that the rise of the friction was irregular, some increases causing fall of speed of the driving motor, the friction falling again on restoring the speed. At 100 deg. C., without waiting for the friction to rise to the stage of stopping the motor, the heat was cut off and the bearing allowed to cool. The return curve did not follow the forward curve; it fell quickly to the minimum value at about 80 deg. C. and was quite smooth. On run-

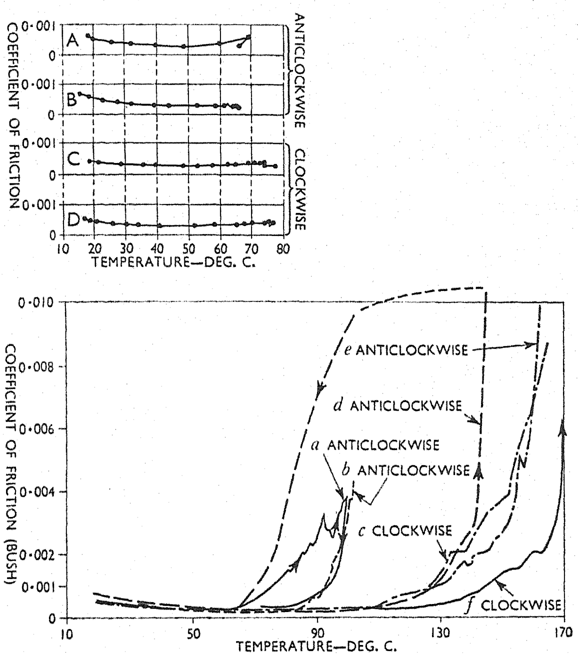

Fig. 2. Friction-Temperature Curves

ning again in the same way, the friction did not rise until 80 deg. C. had been reached, compared with 55 deg. C. in the first run, and the curve followed closely that of the return curve of the first run (curve *b*, Fig. 2); moreover its return curve was closely the same as the forward curve.

The bush was examined after a further run and found to have a large polished area, not quite uniform along the length, extending from 20–30 deg. on the outlet side to 10–20 deg. on the inlet side. The polished areas contracted rapidly and closed before reaching the

ends, because expansion of the projecting ends of the bronze bush had taken them out of contact with the journal during running at the elevated temperatures. This was confirmed by the lack of change of circumferential shape of the bush at position A (Fig. 1). At position B, which corresponded with the widest part of the worn area, the bush was enlarged by about 0·0001 inch (curve *b*, Fig. 3), mainly eccentrically to the crown, on the outlet side. It is thought that the irregularity of the forward portion of curve *a* (Fig. 2) was due to contacts between the journal and bush, and the wearing away of material; and that the

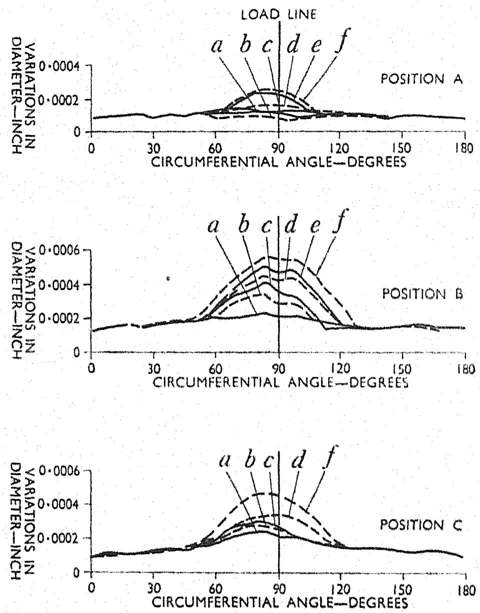

Fig. 3. Circumferential Variations of Diameter of Bush

smoothness of the return curve and the improvement in the subsequent running was due to the presence of a fluid film where previously contact would have occurred.

Further runs were made to see whether any further improvement could be effected, even by allowing the coefficient of friction to rise to 0·010 or over; if anything, however, there was slight deterioration as regards the temperature at which the friction began to rise, and no change in the seizing temperature in the three further runs made. There was a slight further change of shape of the crown (curves *c*, Fig. 3).

Run to Seizure: Clockwise. The bearing was next run in the opposite direction to see whether the eccentric wear at the crown had any effect on the behaviour of the bearing when the direction of rotation was reversed. In the first run the coefficient of friction began to rise at about 70 deg. C., but only slowly; it had only reached 0·0005 at 100 deg. C., the seizing temperature for the other direction of running, and rose moderately smoothly to 0·0065 at 158 deg. C. At this point the rate of rise became very rapid, so the heat was cut off; instead of falling immediately, however, as was usual, the friction retained a high value until the temperature had fallen to 140 deg. C., and then it came down to normal values.

Two further runs were carried out with the same direction of rotation. The friction-temperature curve for the second of these is shown as *c* in Fig. 2. The coefficient of friction did not begin to rise from the minimum until 100 deg. C. had been reached, and a value of 0·008 was reached at just over 160 deg. C. In neither case did the machine stop, and the values of the friction with decreasing temperature were normal. Examination of the bush showed polish of the surface towards the other side of the crown from before; the worn area now extended symmetrically to 30 deg. on each side of the crown, the width decreasing rapidly just at the ends.

Measurements of the bush after these three clockwise runs showed that the enlargement at the crown had become more nearly symmetrical, as shown by curves *d* in Fig. 3. It would thus appear that the modification of shape of the bush to the outlet side of the crown during anticlockwise running, becoming a modification to the inlet side for clockwise running, improved the behaviour of the bearing considerably. This is rather remarkable as the maximum increase of radius was only 0·0002 inch.

It may be noted that, at 100 deg. C., where the rise of friction now begins, the value of ZN/P is 0·8, where Z is the viscosity in centipoises, N is the speed in revolutions per minute, and P is the load in pounds per square inch; in uniform units the value is $1{\cdot}93 \times 10^{-9}$. At 20 deg. C., on starting, the respective values are 15·5 and $37{\cdot}4 \times 10^{-9}$.

Further Runs. Having now made the shape of the crown closely symmetrical, the bearing was run in the anticlockwise direction again. The coefficient of friction (curve *d*, Fig. 2), began to increase at about 90 deg. C. and followed the values for clockwise running to just over 140 deg., and then suddenly increased to a high value above 0·010. The applied heat was immediately cut off, but the friction remained high in spite of the fall of temperature, and only returned to normal values at about 65 deg. C. (see Fig. 2). This behaviour was similar

to that in the first clockwise run, noted above, and was repeated in a further run anticlockwise. As a result of these two runs the enlargement of the crown was only 0·00005 inch at the position B, and practically nil at position C; at position A, however, a local enlargement of the same order as at B appeared, the wear now taking place as near as this to the end of the bush (curves *e*, Fig. 3).

At a later stage a similar anticlockwise run gave a friction-temperature curve, *e* in Fig. 2, very much like that, *c*, for clockwise running, reaching a coefficient of friction of 0·010 at 163 deg. C. A clockwise run following then gave the same coefficient at 168 deg. C. In both cases, on cooling, the coefficient of friction remained high at first in spite of the decrease of temperature. The change of shape of the bush at position B due to these two further runs was only 0·00003 inch, and it therefore seemed that a practically constant state had been reached.

As greater wear would be expected from seizure at higher speed, the next run was made at 300 r.p.m. This would of itself have increased the seizing temperature, however, and to avoid the distortion arising from high temperatures, the load was increased in the same proportion, namely from 500 to 1,500 lb. per sq. in. During the first run, anticlockwise, there was some irregularity of behaviour as the friction increased and the seizing temperature reverted to the value of about 140 deg. C. for this direction of running. In the next, clockwise, run, however, 110 deg. C. was reached before the friction began to rise, and the friction suddenly increased at about 170 deg. C. to stop the machine (curve *f*, Fig. 2). The behaviour in the anticlockwise direction was almost exactly the same, except that seizure occurred suddenly at 156 deg. C. The abrupt increase of friction is characteristic of a higher speed of running. Returning to the initial conditions, a coefficient of friction of over 0·007 was reached at 168 deg. C. in anticlockwise running. A slight improvement was thus found in the runs at higher speed and higher load.

The form of the crown after this running is shown by curves *f* of Fig. 3. The local radius of curvature at the crown at each measuring position is 1·0058 inches, compared with a journal radius of 1·0047 inches and an initial radius of the bush of $1{\cdot}0073_5$ inches.

Conclusions. In a journal bearing test in which the temperature was raised artificially, high point contact between the journal and bush occurred immediately on passing the minimum friction; also the points of contact were eccentric to the centre line. Using a sensitive measuring apparatus the changes of circumferential form of the bush were measured when the bearing was run repeatedly to seizure. With

one direction of running the crown was worn mainly on the outlet side of the centre line. On reversing the direction of rotation this modification of shape, coming now on the inlet side, led to increased safety of the bearing, the seizing temperature under the particular conditions rising from 100 deg. C. to 160 deg. C. Seizure with this direction of rotation rendered the modification of the crown of the bush symmetrical about the centre line, and the safety for the original direction of rotation was improved. The investigation is being continued.

THE LUBRICATION OF THE AUTOMOBILE ENGINE

By C. B. Dicksee *

A discussion on the lubrication of the automobile engine centres largely around the crankshaft and its bearings. The enclosed situation of the principal bearings, together with the high speeds and loadings, necessitates some means of removing the heat generated if destructive temperatures are to be avoided, and the simplest means of doing so is to remove the heated oil film, and replace it by a cool one. The function of the oil supply is thus twofold:—

(1) To provide adequate lubrication and thereby minimize both the friction loss and the heat generated.

(2) To carry away the heat generated and so keep the temperature within practical limits.

Crankshaft Lubrication. Broadly speaking, crankshaft oiling systems may be classed under two headings: (1) the splash system; and (2) the pressure-feed system. With modern speeds and loads the splash system has come to be looked upon as inadequate and the pressure system has become almost universal, although high-speed American designs have recently employed, with success, the trough form of splash system. With careful design the splash system can thus be made suitable for modern conditions and, in view of its freedom from certain defects of the pressure system, it warrants further attention.

At first sight the pressure system would appear to be ideal. By forcing the oil through the bearings, a positive means of removing the heated oil film and substituting a fresh one is provided. By control of the pressure and variations in clearance almost any desired rate of flow can be obtained. The oil is fed directly to the main bearings and the disadvantages of the system mainly arise because the supply to the crankpins has to be taken from the main bearings via passages drilled in the crankshaft.

The oil in the crankcase of the average automobile engine contains appreciable quantities of foreign matter. Forcing large quantities of such oil through the bearings must undoubtedly be detrimental to their life, particularly that of the main bearings which receive all the oil required by the crankpin bearings. With any form of splash system each bearing receives its own quota of oil only. This, on account of the high rate of heat extraction of the oil thrown at a high speed over

* The Associated Equipment Company, Ltd.

the outside surface of the various bearing housings, is much less than that received when pressure feed is employed. Anyone who has examined, after a reasonable period of service, the bearings of two similar engines, the one trough-fed and the other pressure-fed, must have noted the superior appearance of the trough-fed bearings.

Feeding the crankpins via the main bearings reacts unfavourably on the latter in yet another way, in that for an adequate supply of oil to reach the crankpin, a feeding groove must be provided, either partly or wholly around the circumference of the main bearing. This cuts the bearing into two strips which, with the narrow bearings necessary to-day, is, to say the least, undesirable. Also, it reduces bearing area to an extent which is not always appreciated. Some alleviation may be obtained by various devices, but the fact remains that the presence of this groove is undesirable and could advantageously be dispensed with.

The Lubrication of Pistons, Camshafts, etc. The lubrication of all parts other than the crankshaft is commonly provided for by oil thrown off by the shaft. Everything within the crankcase is flooded with oil and the bearings of the valve gear and auxiliary drives are amply provided for. The rush of oil-laden air which follows the pistons on their inward stroke, is more than adequate to ensure cylinder lubrication; in fact provision has to be made, not always successfully, to prevent an excessive amount of oil from passing the piston. The gudgeon pin is occasionally provided with a pressure feed through a hole drilled lengthwise along the connecting rod from the crankpin bearing. This appears to be both unnecessary and undesirable: unnecessary because a small hole drilled in the supporting bosses of the rod and piston will suffice, and undesirable because oil is drained from the connecting rod bearing from the point at which it is most essential that an oil film should be maintained, and also because the oil consumption is likely to be affected adversely.

Oil Grooves. For heavily loaded bearings oil grooves are to be avoided as far as possible. At best oil grooves serve to break the surface up into a number of small areas which must act independently, while in extreme cases they may cause failure by draining away the oil. Also, the loss of area occasioned by the grooves is a matter for consideration. The ideal is a plain unbroken cylindrical surface, the oil being fed in at the point of lowest pressure. For short bearings slight chamfering of the edge of the oil hole usually suffices to ensure effective distribution of the oil, and with bearings of moderate length a short axial groove extending for not more than one-fourth the bearing length should be ample.

The ideal is readily attained with connecting rod bearings which are fed from the interior of the shaft, but with main bearings the circumferential groove necessary for feeding the crankpins when the pressure system is used, prevents the ideal from being attained. If the main bearings could be given fairly generous width, the presence of the groove would matter little, but under modern conditions it is a nuisance.

For lightly loaded bearings such as camshaft bearings, etc., oil grooves have their advantages as no positive feed pressure is provided. Some form of catchment delivering oil to a suitable groove ensures that oil is fed into the bearing. A single spiral groove turning through about 90 deg. in the length of the bearing in a direction such that the rotation of the shaft tends to drive the oil along the groove through the bearing will usually be adequate.

Oil Pressure and Oil Supply. The working oil pressure is usually fixed at some figure at which test-bed experience has shown that the engine will operate satisfactorily. At high speeds, centrifugal action upon the oil within the passage drilled in the crankshaft exerts an important influence upon the supply of oil, not only to the crankpin, but to the main bearings also. Before the oil can reach the crankpin, it must first travel radially inwards against centrifugal pressure. On the crankpin side a similar action takes place, but here the centrifugal action usually tends to promote, instead of oppose, the oil flow. The centrifugal action at the inlet end of the passage is thus opposed to that at the outlet end. The extent of this action may be readily determined from the usual formula for determining centrifugal force and for an oil having a specific gravity of 0·9 the expression reduces to $P=0{\cdot}462N^2R^2\times10^{-6}$, where P is the resultant pressure in lb. per sq. in., N is the speed of the shaft in r.p.m., and R is the radial distance, in inches, of the opening from the centre of the shaft. Thus a crankshaft running at 4,000 r.p.m. and having a throw of 1·5 inches and a journal and crankpin diameter of 1·5 inches will have a centrifugal pressure at the entry to the main journal of 4·16 lb. per sq. in. and at the outlet from the crankpin of 37·4 lb. per sq. in. if drilled at the extreme surface of the crankpin. If drilled at right-angles to the throw, the pressure will be 16·6 lb. per sq. in., and when drilled on the side of the pin towards the centre of the shaft 4·16 lb. per sq. in. Usually, but not invariably, the radius of the opening in the crankpin is greater than that in the main journal and the net result is a tendency for the oil to flow towards the crankpin. The combination forms a powerful centrifugal pump which exerts a profound influence upon the oil flow and by varying the position of the crankpin opening a considerable measure of control over the oil flow may be exerted.

The opposing action of the two sides of the system places the oil under tension and in extreme cases an interruption of the flow may occur owing to the separation of the oil column. To avoid this the pressure in the feed groove in the main bearing should, at least, be equal to the centrifugal pressure on that side. The pressure on the feed side will then never fall below atmospheric pressure and the separation of the column on the feed side, and therefore an interruption of the supply, will be avoided.

This centrifugal action is such that the flow to the crankpin increases rapidly with speed even without any increase in pressure of the supply. This indicates that care is necessary to ensure an adequate supply of oil in order that the main bearing may not be starved by the crankpin. Although the oil reaches the main bearing first, it does not actually pass through the main bearing before reaching the crankpin, but both are fed from a common canal—the circumferential groove in the main bearing—and if the crankpin tends to drain this canal faster than it is being filled, oil, instead of being forced out into the bearing, will tend to flow from the bearing to the groove, so that air will flow into the bearing and failure may result. Should an appreciable quantity of air reach the feed groove, it will ultimately enter the crankshaft passage to cause a separation of the oil column and interrupt the supply to the crankpin. This will be followed by a restoration of the pressure in the feed groove, and of the supply to the main bearing, but if the pressure does not equal, or exceed, the centrifugal pressure at the journal, 4·16 lb. per sq. in. in the example, the flow to the crankpin will not be resumed and failure must follow.

The absolute necessity of maintaining at all times a pressure in the feed groove in excess of that due to the centrifugal pressure is thus manifest. In this connexion, it is not always realized that the hydraulic gradient along the supply pipes is considerable, and varies greatly with the viscosity of the oil and, therefore, its temperature. Under such conditions a serious shortage could result if the engine is run at high speeds before the oil has thinned out sufficiently to allow adequate supplies to reach the bearings most remote from the pump.

In many instances the feed groove does not extend for more than part of the bearing circumference. This results in an intermittent supply to the crankpin and has the advantage that when the feed is cut off from the rod, the pressure has a chance to build up in the groove. With this arrangement, a continuous feed to the rod may be obtained by feeding from bearings on each side of the throw.

The extent of the registration between the crankpin and the oil supply bears directly upon the oil supply and provides a further means of controlling the quantity of oil flowing. The flow is also governed

by the physical dimensions, speed, and clearance of the bearing. Of these three, only the clearance is available for oil control. It is, however, a control which varies on account of wear.

Oil Filtration. The effective filtration of the oil would undoubtedly be beneficial. Relatively few engines are fitted with anything beyond a gauze strainer around the suction pipe, an arrangement which serves rather to protect the pump from large particles than the bearings from abrasive matter. The chief difficulties in the way of the provision of an effective filter are space and cost. The rate at which the oil is circulated necessitates a large filter if the filtration is to be effective or if the filter is to remain serviceable for a reasonable period. Alternatively, a portion of the oil may be bypassed through a filter, an arrangement which, although it maintains the oil in a better average condition than when no filter is fitted, does not adequately protect the bearings should abrasive matter be present. For fleet operators, probably the best solution is to change the oil frequently and reclaim the drainings, using them for "topping up", new oil being used for refilling.

The Lubricating Oil and Sump Operating Temperatures. The automobile engineer has to use the same oil to lubricate both cylinder and crankshaft. The oil has therefore to be suitable for widely different conditions. It is subjected to extremes of temperature under strongly oxidizing conditions, it is diluted with fuel, contaminated by products of combustion, and those produced by its own decomposition, as well as by dirt and metallic particles resulting from wear. Under these conditions it must lubricate the cylinder and piston at temperatures of 250 deg. C. and even higher, and at the same time must be suitable for lubricating bearings which are running at high speeds and carrying heavy loads. A great deal is thus demanded from the oil and though great improvements have taken place, there is still room for many more.

The maximum safe temperature of the sump oil depends largely upon the bearing materials employed. With tin base materials of the ordinary type oil temperatures in excess of 100 deg. C. cannot be tolerated for more than occasional short periods. The strength of the material decreases rapidly as the temperature increases beyond this figure and even at 100 deg. C. the strength is seriously diminished. With bearings of the copper-lead type higher temperatures can be used without danger of the bearing lining breaking up, and materials of a softer nature have recently appeared for which the same claim is made. Under service conditions an oil temperature of 80 to 85 deg. C. is commonly

reached with heavy commercial vehicles and the bearings themselves are some 30 deg. C. hotter than the sump oil. The need for a bearing material having ample resistance at such temperatures is evident.

Under a given set of conditions of engine speed and crankcase cooling the equilibrium temperature reached by the oil is very nearly the same when the engine is motored from an external source of power as when run at full throttle. This means that very little combustion heat reaches the oil and that the heat in the oil is due, in the main, to bearing friction. The work lost by bearing friction largely depends upon oil viscosity, so that an engine lubricated by a heavy oil will have a greater friction loss and therefore reach a higher oil temperature than if a lighter oil were used. Subject, therefore, to the viscosity being adequate at operating temperatures, a light oil is to be preferred to a heavy one, especially as under easy conditions and low temperatures the viscosity of the light oil will be very materially less than that of the heavy one with a corresponding reduction in engine friction loss.

THRUST BEARINGS FOR STEAM TURBINE MACHINERY

By R. Dowson, B.Sc., M.I.Mech.E.*

Tilting-Pad Thrust Bearings

The modern tilting-pad single-collar thrust bearing is the outcome of twenty-five years of development by a number of makers, and in consequence of accumulated experience the doubtful or bad practices in design have been gradually eliminated. There are to-day many points of general agreement between one design and another, and few important differences. It is the object of this paper to discuss the guiding principles very briefly and to draw attention to points of agreement and difference in practical application.

(1) *Lubrication.* The factors involved are discussed below:—

Geometrical Proportions. The present practice of seven leading steam turbine manufacturers is summarized in Table 1 (pp. 77–9), which will be made clear by reference to Fig. 1.

There is general agreement that "square" pads should be used (B=L) with rounded corners. The ratio thickness to circumferential length of pad (T/L) is usually about 1/3. If the pads are too thin they are mechanically weak, whilst if they are too thick, instability and failure of the oil film may result. The radial width B of the pads is determined by the total surface required to sustain the load. If it is made too wide, then with square pads the number of pads becomes too small to secure proper tilting in the direction of sliding. In practice, the minimum number of pads is about 10 and the angle θ subtended at the centre not more than about 27 deg. The total pad surface is about 80 per cent of the complete annulus, gaps g between pads being necessary in order to secure individual lubrication.

Eccentricity of Tilting Axis. The degree of eccentricity of the tilting axis of the pads from their geometrical axis is a matter in which there is some difference in practice. In marine work where the direction of rotation has to be reversible it is necessary to adopt central pivoting, i.e.

$$e/L=(b-a)/2L=0 \qquad \ldots\ldots\ldots \quad (1)$$

Such centrally pivoted bearings function satisfactorily, though the load that can be sustained is less than that permissible when eccentric

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pivoting is adopted. On the other hand, too great an eccentricity is to be avoided because it results in a very thin oil film at the trailing edges of the pads, and any fine grit that may be and generally is present in the oil scores these edges.

An eccentricity of 16·67 per cent, or one-sixth of the length L of the

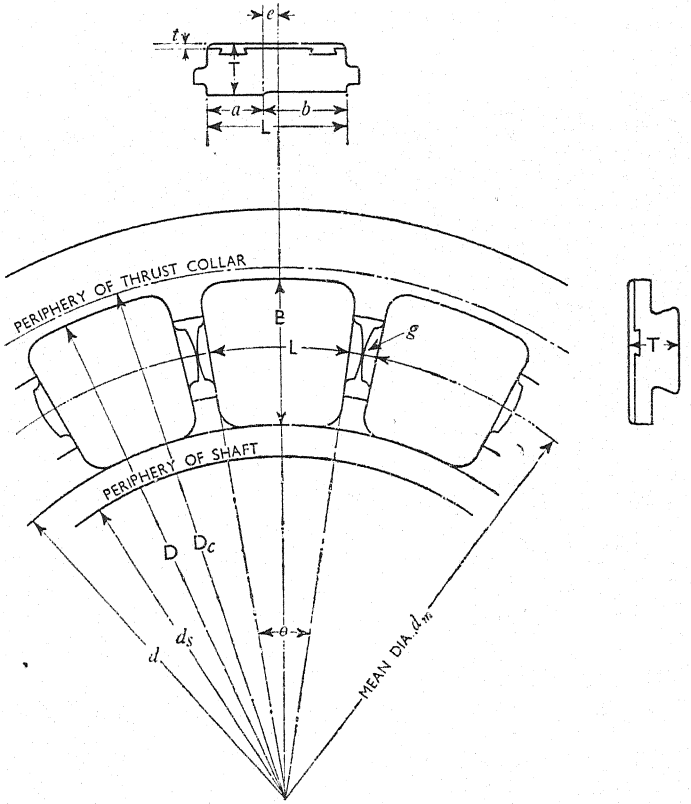


Fig. 1. Typical Design of Tilting Pads

Plan looking on faces of pads.

pad, appears to be the maximum adopted, most manufacturers preferring a considerably lower value.

Materials Used and Finish of Pad Surfaces. The sliding surface of the pads in all instances is a tin-base whitemetal, generally of tin, antimony, and copper, but the minimum thickness adopted varies from $\frac{1}{16}$ to $\frac{1}{8}$ inch. The finish of the surfaces is effected by scraping, the pads being made in the form of a complete ring of steel or brass

which is white-metalled and accurately machined before cutting up The pads are then scraped to a gauge.

Oil Quality and Viscosity. The oil used for thrust bearings is the same as that used for the journal bearings of a turbine, the properties required for the two purposes being similar. A wide choice of viscosity is permissible, oil being employed conforming to British Standard Specification No. 489 within the range of medium to extra-heavy grades.

Oil Quantity. Because of the heat generated the quantity of oil required for high-speed thrust bearings is considerably greater than that necessary to maintain the oil film. The amount supplied is adjusted usually by restricting the outflow so that a specific temperature of the oil, measured at discharge, is not exceeded. This temperature is not necessarily that of the sliding surfaces, or of the oil between those surfaces, but merely a working oil discharge temperature found from experience to be satisfactory when all precautions have been taken to secure complete and continuous lubrication. The heat generated in a thrust bearing may be estimated with fair accuracy and the quantity of oil required can then be calculated for any given temperature rise. The maximum temperature rise adopted in practice is about 25 deg. F.

Total Oil Clearance. The total oil clearance c permitted for the complete thrust bearing (both sides) is never less than 5–8 mils (1 mil $=0{\cdot}001$ inch). For convenience of calculation manufacturer d adopts a working formula $c=0{\cdot}82\sqrt{\mathrm{VL}}$ for the required value of the clearance in mils, V being the mean surface velocity and L the circumferential length of each pad. If the clearance is increased the frictional losses are reduced, but there is clearly a limit to the free axial movement of the shaft that can be allowed. Having regard to both these considerations the above formula gives values for the clearance which are satisfactory and consistent. Manufacturer f adopts a practical rule in which the total oil clearance allowed depends directly on L.

Calculation of Friction Losses. The losses are given by AVS/J =B.Th.U. per sec., where A is the total pad surface and S the shear resistance per unit area. This involves a knowledge of S for each side of the thrust collar. Methods of calculation vary with different manufacturers, but usually are based on Michell's and other published researches.* The shear resistance is usually determined from the relation

$$\mu=\frac{\mathrm{S}}{p}\propto\sqrt{\frac{\mathrm{ZV}}{p\mathrm{L}}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

* Boswall, R. O. "The Theory of Film Lubrication", Longmans, Green and Co., 1928; Michell, A. G. M. *Zeitschrift für Mathematik und Physik*, 1905, "Lubrication of Plane Surfaces"; Howarth, H. A. S. Trans. A.S.M.E., 1935, vol. 57, MSP-57-2.

where Z is the viscosity of the oil, p the pad pressure, and μ the coefficient of friction.

The total oil clearance c is the sum of the individual oil film thicknesses t_1 and t_2 on either side of the thrust collar, and the relation between t and p is

$$\frac{t}{L} \propto \sqrt{\frac{ZV}{pL}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

This has to be solved for both sides of the collar in order that the appropriate values of p may be substituted in equation (2). There is no direct way of approaching the analysis, but sets of curves can be constructed from which the losses may be evaluated for known values of V, p, and L. Such charts form a convenient reference for design purposes.

Arrangements for Admission, Circulation, and Drainage. The thrust bearing may be arranged: (1) entirely separate from a journal bearing; (2) adjacent to a journal bearing; or (3) within it. In all three arrangements a separate oil supply is provided to each side of the thrust collar and particular precautions are taken to ensure continuous bath lubrication. Usually the oil is admitted under a pressure of about 10 lb. per sq. in. to the inner periphery of the pads in a manner ensuring even distribution all round, and the oil is drained away from the top of the housing. The rotating thrust collar tends to pump the oil through the bearing and some restriction of the outlet duct is necessary in order to prevent too free a discharge, which might prevent oil from reaching all parts of the sliding surfaces. When the entire assembly is totally immersed in oil as described above, it is not usual to fit deflector plates or scraper rings, but a baffle plate is sometimes placed around the periphery of the thrust collar in order to keep the oil drains from each side of the collar separate. This enables one side of the thrust bearing to be supplied with more oil than the other, the areas of the drain holes on either side of the baffle being suitably adjusted for the purpose.

(2) *Surface Velocity.* The surface velocity permissible is the limiting factor in high-speed thrust bearing design. The mean diameter of the pads is usually taken as the diameter which divides each pad into two equal areas. The velocity at this mean diameter generally does not exceed 170 ft. per sec., although a maximum of 216 ft. per sec. has been found satisfactory. The peripheral velocity is, of course, considerably greater and limits the permissible outer diameter of the pads. The inner diameter of the pads is determined by the minimum diameter of shaft or journal considered necessary from other considerations, and

so the total area of the pads that can be provided is limited. In practice $\lambda=d/D$, the ratio of the inner and outer diameter of the pads, lies between 0·6 and 0·8, while the factor b, or the ratio of the total surface of the pads to the complete annular area, is about 0·80. Calling $\bar{A}$ the total area of the circle of diameter D, and A the total area of all the pads on one side of the collar, then if $b=0{\cdot}80$ and $\lambda=0{\cdot}70$,

$$\frac{A}{\bar{A}}=b(1-\lambda^2)=40{\cdot}8 \text{ per cent} \quad . \quad . \quad . \quad . \quad . \quad (4)$$

Where a heavier load has to be sustained in one direction than in the other, it may be arranged that the greater of the two shall be towards the end of the shaft away from the blading. This permits the shaft to be stepped down to a smaller diameter on that side, so that greater pad area can be provided without exceeding the permissible maximum diameter.

Double-Collar Thrusts. Occasionally double collar tilting-pad thrust bearings have been made in order to carry a heavy load in one direction. Satisfactory operation has been obtained but it is doubtful whether the two collars really share the thrust equally, in spite of all precautions taken to bring this about. Both British and American examples of this practice can be found.

(3) *Pressure on Pads, Total Load Carried, and Mechanical Strength.* The mechanical strength of thrust bearings is a matter of prime importance because it is necessary to support a total load as well as a unit pressure. Although with perfect lubrication and alignment the pads will sustain much greater pressures than those given in Table 1, practical considerations make a certain prudence advisable. The total load on the bearing may be larger than that anticipated, so that the nominal pad pressure may in fact be exceeded. Excessive total load may cause distortion of the thrust bearing in some of its components and this may disturb the alignment.

(4) *Alignment.* In British practice it is customary to rely on accurate workmanship to secure even loading of all the pads when in service, and as a rule no special devices are adopted for distributing the load automatically in the event of some pads becoming more heavily loaded than others. A spherical seating of the thrust housing is, however, adopted by some manufacturers.

(5) *Axial Adjustment.* In all but small thrust bearings it is usual to provide some form of adjustment operated by hand, so that the axial position of the turbine rotor can be set in a predetermined relation to

TABLE 1. TILTING-PAD THRUST BEARINGS FOR STEAM TURBINE MACHINERY

* Bearings of the Michell Bearing Company.
† p=mean pressure, pounds per square inch; r_m=mean pad radius, inches; N=speed in revolutions per minute; θ=angle btended per pad, radians; A=total pad area, square inches.

No.	Reference letter of maker	*a*	*b*	*c*	*d*	*e*	*f*	*g*
1	Pressure on pads, lb. per sq. in.:—							
	Normal .	Up to 150	*	Up to 250	Up to 150	Up to 300	Up to 150	Up to 200
	Maximum	350		300	500	500	500	400
2	Surface velocity V at mean dia., ft. per sec.:—							
	Normal .	50–120	*	130–170	150	60–140	100	120
	Maximum	187		216	172	150	190	160
3	Ratio of breadth to length of pad, B/L	1/1	*	1/1	1/1	1/0·93 to 1/1·15	1·1/1·0	1/1·7
4	Ratio of thickness to length of pad, T/L	1/3	*	1/2·0 to 1/2·5	1/3	1/2·48 to 1/3·75	1/3	1/4
5	Minimum thickness of whitemetal, inches, t	$\frac{1}{16}$	*	$\frac{1}{16}$	$\frac{1}{16}$	$\frac{3}{32}$	$\frac{1}{8}$	0·06
6	Method of finishing pad whitemetal surfaces	Scraping	*	Scraping	Scraping	Scraping	Scraping	Scraping

TABLE 1, concluded

o.	Reference letter of maker	*a*	*b*	*c*	*d*	*e*	*f*	*g*
7	Percentage composition of whitemetal used	Sn 86 Sb 8·5 Cu 5·5	*	Sn 76 Sb 10 Cu 6 Pb 8	Sn 83·3 Sb 8·33 Cu 8·33	Sn 83 Sb 9 Cu 8	Sn 77 Sb 14 Cu 9	Sn 85 Sb 8·5 Cu 6·5
8	Amount of eccentricity *e*/L of pivots from centre line of pads	10 per cent =L/10 *a*/*b*=2/3	*	16 per cent =L/6·25 *a*/*b*=1/1·94	5 per cent =L/20 *a*/*b*=9/11	16·7 per cent =L/6 *a*/*b*=1/2	10 per cent =L/10 *a*/*b*=2/3	Zero
9	Total oil clearance, mils.— Minimum Normal . Maximum	12 14 20	*	8 — 12	$0{\cdot}82\sqrt{VL}$ — —	14 — 40	4 to 5 per 1 in. of L —	10 — 12
0	Quantity of oil in cu. ft. per min. is made sufficient for	20 deg. F. temp. rise (from 12)	*	25 deg. F. temp. rise (from 12)	15 deg. F. to 25 deg. F. (from 12)	—	15 deg. F. temp. rise (from 12)	15 deg. F. to 25 deg. F. (from 12)
1	Viscosity of oil	Redwood 50 cu.cm., 75 sec. at 140 deg. F. Max. 90 sec. (direct coupled sets)	*	Redwood 50 cu.cm., 90 sec. at 140 deg. F.	Redwood 50 cu.cm., 90 sec. at 140 deg. F.	Redwood 50 cu.cm., 90 sec. at 140 deg. F.	Redwood 50 cu.cm., 70 sec. at 140 deg. F.	Redwood 50 cu. cm., 70–120 sec. at 140 deg. F.
2	How frictional losses are estimated	† $\text{kW.}=\frac{4{\cdot}16}{10^8}\,\frac{p^{\frac{1}{2}}r_m N^{\frac{3}{2}} A}{\theta^{\frac{1}{2}}}$	*	Michell curves	From curves based on published researches	From curves based on published researches	Michell curves	From curves based on published researches

13	Seat housing used	Usually spherical	Spherical	Spherical on larger sizes	Not spherical	Not spherical	Not spherical	Not spherical
14	Arrangement of pads on both sides of collar	May be unsymmetrical when end pressure is large in one direction	May be unsymmetrical when end pressure is large in one direction	Symmetrical	May be unsymmetrical when end pressure is large in one direction	May be unsymmetrical when end pressure is large in one direction	Symmetrical	May be unsymmetrical
15	Provision for equalizing loading automatically	Pads carefully levelled during manufacture + spherical seating	Pads carefully levelled during manufacture and mounted on solid support	No special provision	Pads carefully levelled during manufacture and mounted on solid support	Complete ring scraped to two surface plates before parting	No special provision	No special provision
16	Thrust bearing separate or combined with a journal bearing	Usually combined	Combined	More often combined	Combined	Both are used	Mostly separate	Separate but close to a journal bearing
17	Solid or flexible coupling between turbine and alternator	Flexible in bending, stiff axially	Flexible jaw type or semi-flexible bellows type	Flexible	Flexible	Solid	Mostly flexible	Flexible for direct coupled sets
18	If flexible, alternator rotor end thrust taken by	Coupling	Limiting plates if flexible jaw coupling used	Tilting-pad thrust bearing for large machines, collar thrust bearing for small machines	Tilting-pad thrust bearing	—	Limiting plates in coupling or by thrust bearing	Generally collars for location

the casing. Since the total thrust carried may be several tons, it is necessary that the adjusting gear shall be robust and rigid. Usually the thrust bearing housing is made in two halves with the joint on the horizontal centre line. The two halves are keyed and bolted together and the complete housing is moved backwards and forwards by means of either a worm and wormwheel linkage, or a screwed nut round the outer surface of the housing with wormwheel teeth cut on its outer periphery. Generally a single worm shaft operated by hand from

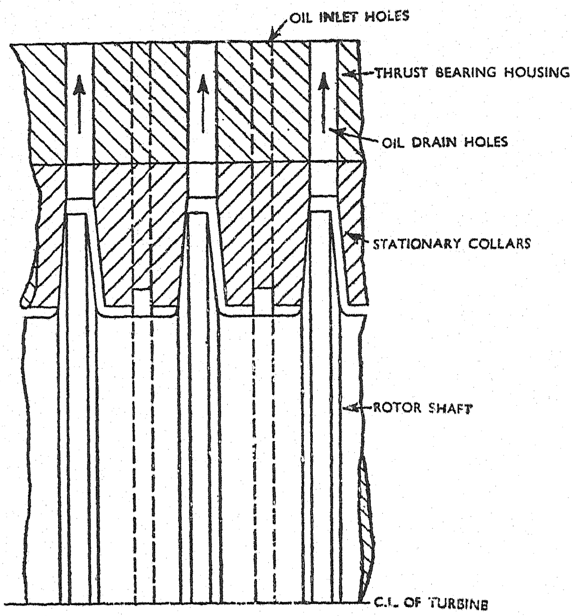


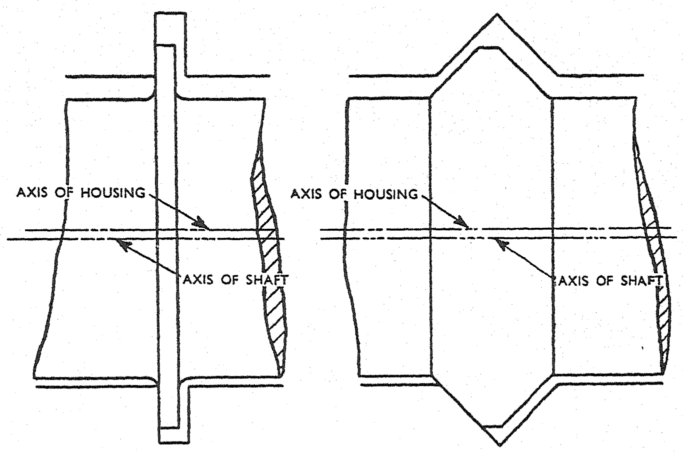
Fig. 2. Bevelled Collar Thrust Bearing

outside the turbine enables the thrust bearing to be adjusted between two limiting liners.

(6) *Alternator Rotor Thrust*. Tilting-pad thrust bearings are also adopted in some instances to locate the alternator rotor, especially when a flexible coupling of the claw type is used to couple it to the turbine. Such couplings in practice may transmit considerable load in an axial direction. Some makers prefer to use a form of coupling which is flexible in bending but stiff axially, so that the alternator rotor may be located by it, a certain freedom being allowed by the use of limiting plates.

Collar and Related Types

Plain Collar Type. Although the tilting-pad type of thrust bearing generally has displaced the collar type, the latter is still used by some manufacturers for small turbines. Such bearings may be brass- or whitemetal lined. In the original design of multi-collar bearing there were two separate halves, the top half being adjustable in an axial direction. There was, however, (*a*) insufficient provision for getting the oil on to the working surfaces, and (*b*) inadequate circulation, so that not more than 6–10 lb. per sq. in. could be carried with certainty.



a Parallel collar bearing

b Bevelled collar bearing

Fig. 3

In modern practice:—

(1) Oil is fed to the inner edges of the fixed collars.
(2) Segmental fixed elements are used, cut away so as to allow oil to be drawn in between the sliding surfaces or provided with radial oil grooves.
(3) Oil passages are carefully designed so as to ensure circulation and removal of all heated oil.

With these improvements a much better performance is now attainable.

Bevelled Collar Type. The bevelled collar thrust bearing (Table 2, col. 3, and Fig. 2) is superior to the parallel multi-collar type in load-

TABLE 2. COLLAR TYPE THRUST BEARINGS FOR STEAM TURBINE MACHINERY

No.	Reference letter of maker	*b*	*f*
	Type of collar	Parallel collar	Bevelled collar
1	Pressure of pads, lb. per sq. in.:— Normal .	20	60
	Maximum	30	150
2	Surface velocity V at mean diameter, ft. per sec.:— Normal .	60–80	120
	Maximum	178·5	140
3	Minimum thickness of whitemetal, t	—	$\frac{1}{8}$ inch
4	Method of finishing whitemetal surfaces	Scraping	Scraping
5	Percentage composition of whitemetal used	Sn 83·3 Sb 8·4 Cu 8·3	Sn 77 Sb 14 Cu 9
6	Total oil clearance, mils.	—	1 per inch dia.
7	Quantity of oil, cu. ft. per min.	$\frac{1{\cdot}55}{10^6} nN(r_1^3 - r_2^3)$ *	From losses and 15 deg. F. temp. rise
8	Viscosity of oil	—	Redwood, 50 cu. cm., 85 sec. at 140 deg. F.
9	How frictional losses are estimated	$\text{H.p.} = \frac{0{\cdot}13}{10^6} W \omega r_m$ †	$\text{kW.} = \frac{1{\cdot}2}{10^6} nND^3$ ‡
10	Seat housing used	Spherical	Not spherical
11	Provision for equalizing loading automatically	No special provision	No special provision
12	Thrust bearing separate or combined with a journal bearing	Separate	Separate
13	Coupling to gear	—	Thrust bearing in gear box
14	Coupling to generator	—	Flexible

* n = number of collars; N = speed, revolutions per minute; r_1, r_2 = outer and inner radius of collars, inches.

† r_m = mean radius of collars, inches; ω = angular velocity, radians per second; W = total thrust, pounds.

‡ D = outside diameter of collars, inches.

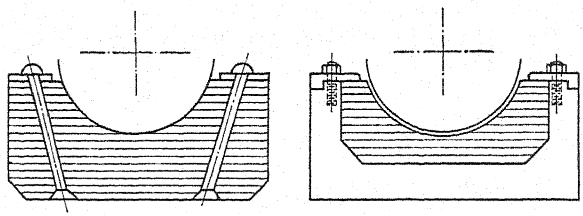
carrying capacity. The capacity is stated to be about four times as great. Care is taken to ensure proper circulation of the oil. Thus the oil is (1) fed through radial holes to the inner periphery of the fixed elements, or (2) supplied to an oil well running the full length of the bearing so that it must reach all sliding parts, and (3) drained away from each collar by a common duct at the top of the housing. The greater effectiveness of the lubrication is probably due to the slight working eccentricity of the conical surfaces (compare Figs. 3*a* and 3*b*) which reproduces in a reasonably satisfactory manner the oil-film generating property of a normal journal bearing.

PROPERTIES AND PERFORMANCE OF BEARING MATERIALS BONDED WITH SYNTHETIC RESIN

By G. R. Eyssen*

According to their interior structure and design, the range of synthetic resin bonded bearing materials can be divided into three groups, each of which shows variations due to its particular applications. Group 1 comprises bearings made from plain or graphitic moulding powders. Group 2 covers bearings made from laminated products, and Group 3 comprises mouldable bearing materials based on felts of impregnated non-orientated fibres.

Group 1 covers some less important materials which are manufactured in considerable quantities and comprise small bushes made



a *b*

Fig. 1. Bearing Composed of Laminated Materials Bolted together

from more or less graphitic moulding powders which are run dry or are lubricated by almost any liquid according to the nature of the assemblies for which they are used. These bushes are very brittle and fragile and their application is very limited, being chiefly for carburettors, spindles of textile machinery, and gland rings. There is now a tendency to replace these moulding materials by those of Group 3.

The most important group of bearing materials bonded with synthetic resin consists of laminated bearings based on specially woven textile fabrics but occasionally made from laminated paper products. Figs. 1–4 illustrate the development of designs for such bearings. Fig. 1 shows an out-of-date type, while Figs. 2–4 illustrate designs which are satisfactory in use All these bearings operate most efficiently with water alone for both lubricating and cooling. Should water not be available or desirable for any reason, almost any liquid

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* Hydro-Plastics, Ltd.

(except strong alkalis) can be used, and although it might be recommended sometimes to add a lubricant, e.g. tallow or suitable emulsions, to the water, these bearings do not require oil or grease. The elimination of oil and grease is a big advantage, especially as

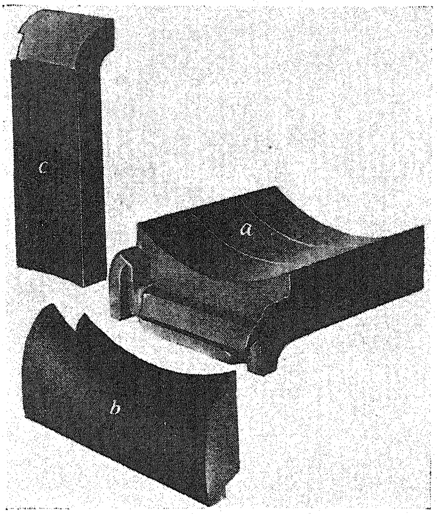


Fig. 2. Block Bearing *a* with Collar *b* and Top Segment *c*

the main application of these bearings is heavy-duty work in rolling mills and similar machinery.

The average physical and mechanical properties of a good laminated bearing material are as follows:—

TABLE 1. PROPERTIES OF LAMINATED MATERIAL

Specific gravity	1·34–1·38
Tensile strength	9,000–12,000 lb. per sq. in.
Young's modulus (approx.)	1,000,000 lb. per sq. in.
Brinell hardness No.	38–42
Compressive strength:—	
At right-angles to laminæ	40,000–45,000 lb. per sq. in.
Parallel to laminæ	20,000–24,000 lb. per sq. in.
Water absorption: 24 hours' immersion, test piece $2\times2\times\frac{1}{2}$ inches	0·5–0·8 per cent
Oil absorption	negligible
Coefficient of friction (lubricated with water only)	0·015–0·025
Thermal conductivity	0·15–0·20 kg.-cal. per metre per hr. per deg. C.

A valuable feature is that laminated bearings reduce the wear of shafts and roller necks. The wear of the bearing surfaces themselves is negligible compared, for example, with that of bronze bearings. Table 2 gives values of the coefficient of friction for laminated bearings compared with metal bearings. Reports of prolonged running verify the conclusions which can be drawn theoretically from the figures of

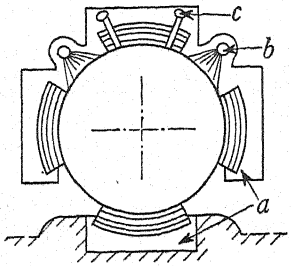


Fig. 3. Four-piece Bearing for a Rolling Mill

a Frames. *b* Water-cooling system. *c* Pressure lubrication.

physical and chemical properties. Furthermore the performance of laminated bearings for heavy-duty work is superior to that of metal bearings, because they are much more resistant to shocks caused by the starting and the feeding of machinery; and because of the ease with which they can be replaced and cleaned owing to their considerably lighter weight and simpler design.

TABLE 2. AVERAGE COEFFICIENTS OF FRICTION FOR LAMINATED BEARING MATERIALS AND BRASS

Lubricant	Brass	Laminated material on cotton base
Dry	—	0·25–0·42
Grease	0·10	0·14–0·12
Oil	0·06	0·07
Emulsion	0·085	0·05–0·08
Water	0·09–0·2	0·03–0·07

The different figures for the synthetic material are due to the various makes, conditions, and the after-hardening effects of time and temperatures. The figures were obtained under the same conditions and these were not very favourable to synthetic bearings.

Almost the only point requiring special attention in the application of laminated bearings is their extremely low thermal conductivity; adequate water cooling must therefore be provided to prevent charring of the bearing surface. Charring, however, is not followed by scoring of the shaft. Figures for thermal conductivity are given in Tables 1 and 2; the thermal conductivity of bearing metals varies from 45 up to 100 kg.-cal. per metre per hr. per deg. C.

A considerable reduction of power consumption and, or alternatively, a higher output under the same conditions can be achieved. The

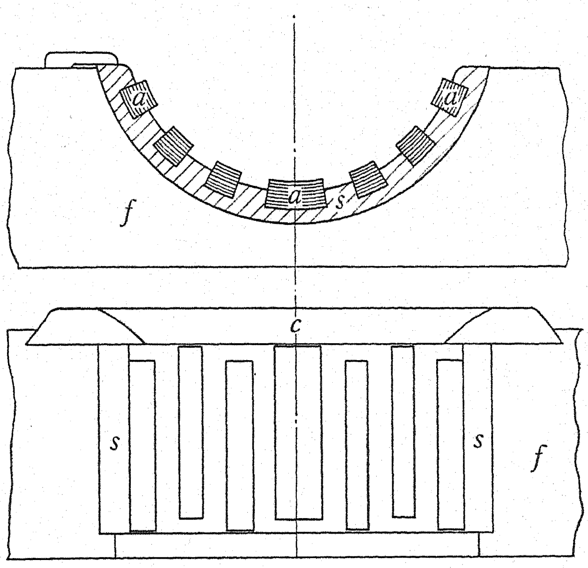


Fig. 4. Bottom Half of a Laminated Bearing with Collar

The bearing strips are inserted in a bent steel plate *s*

a Bearing strips. *c* Collar. *f* Frame. *s* Steel plate.

economy obtained is stated variously as a reduction in power consumption of 25 to 50 per cent, or 6–10 times longer life and 25 per cent increase of output for roller mills, for example.

The economies obtained by the use of laminated bearings for heavy-duty work suggested that similar advantages could be obtained by using such bearings for smaller machinery and especially for medium- and high-speed work. The difficulties to be overcome are mostly due to the low thermal conductivity and the high costs of machining laminated material into bearings of smaller size; moulding into the shapes required is not possible because laminated materials cannot be

treated like moulding powders. Special requirements, however, where oil cannot be used or where complicated designs, packings, and glands are necessary to protect metal bearings from corrosive liquids, have

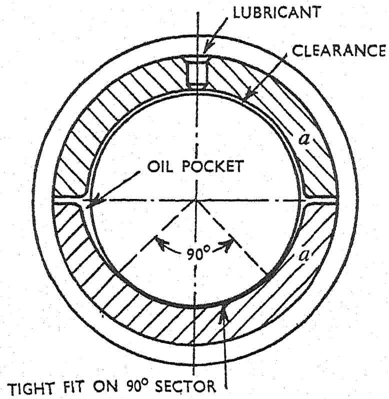


Fig. 5. Split Bearing Lubricated with Oil Emulsion

a Bearing material consisting of phenol-formaldehyde resin, cellulose fluff, and chlorinated diphenyl.

opened up an important new field of application. Higher costs may be counterbalanced by the damage which might be caused by oils and greases (as in textile machinery and pigment grinders) and considerable economy and simplification of design may be achieved by using such bearings in the wide field of chemical engineering, especially where corrosive liquids or atmospheres have to be encountered. In these

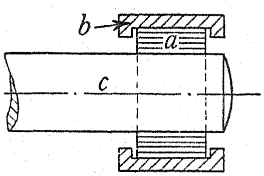


Fig. 6. Laminated Bushes fixed to Shaft

a Laminated bushes. *b* Steel bearing surfaces. *c* Shaft.

cases laminated bearings are almost inert. In Germany laminated bearings are also used in combination with oil for trucks and other applications. Fig. 5 illustrates the design of such a bearing, which is however, not of great interest for countries which do not lack bearing metals.

Another design (Fig. 6) indicates a method of overcoming the low thermal conductivity of such bearings when used for high-speed shafts of smaller diameter. It is obvious that if the heat is not dissipated by the framework, the temperature rise on the shaft will be very high unless efficient water cooling is provided. Whilst cooling is not difficult in rolling mills, it might become very awkward for small machines. A way of overcoming this disadvantage for some special applications is to fit on the shaft laminated bushes which run on steel inserts in the bearing housings. The heat produced between the gliding surfaces will be easily dissipated through the metal housings into the framework, and thus up to 60 per cent higher speed can be safely applied under the same conditions.

Modern trends in the manufacture of moulding materials have created a product which can be actually moulded and which in its physical properties ranges between the fragile powder mouldings and the extremely tough and strong laminated materials. This material, which can be widely varied and adapted to special requirements, forms the basic material of Group 3. It is based on an interior felt-like structure which formerly could be only partly obtained by chipping precondensed laminated sheets and shaping these chips in a second operation under heat and pressure, but the flow characteristics were rather poor. The new material, known in Germany as "Wolff- and Pe-Te Presstoff" is made by various methods, by which a product of feltlike interior structure is obtained by impregnating fibrous fillers, e.g. cellulose fluff, linters, etc., just when felt formation has started —a process well known in papermaking. The resin is generally precipitated from a thin aqueous solution (e.g. sodium hydroxide) on and in the felt-forming fibres.

In Great Britain some progress has been made in the manufacture of such materials by using emulsified resins for the treatment of such

TABLE 3. AVERAGE PHYSICAL AND MECHANICAL PROPERTIES OF GROUP 3 MATERIALS

Specific gravity	1·38–1·42
Tensile strength	6,000–10,000 lb. per sq. in.
Young's modulus (approx.)	1,000,000 lb. per sq. in.
Brinell hardness No.	30–40
Compressive strength	25,000 lb. per sq. in.
Water absorption (immersion for 24 hours)	0·3–0·8 per cent
Coefficient of friction (lubricated with water)	0·02
Thermal conductivity	0·26–0·30 kg.-cal. per metre per hr. per deg. C.

fibres or fibrous felts. Table 3 records some physical properties of moulding materials made on the above described principle. These felts, not only of coated but also of filled fibres, flow easily even in quite intricate moulds, the final product showing a homogeneous non-orientated structure. The material, however, in order to preserve the fibres, is not marketed in the form of a powder but in slabs, lumps, sheets, rods, or tapes.

Having a material of such novel physical properties it was logical to use it for moulded bearings. The method of preparation led automatically to the next stage of development, which was the dispersion of very finely ground graphites or colloidal graphites. The result was a bearing material which was suitable for a wide field of applications within the limits set by the properties of the material itself and the conditions imposed by load and speed. Fig. 7 indicates the load and

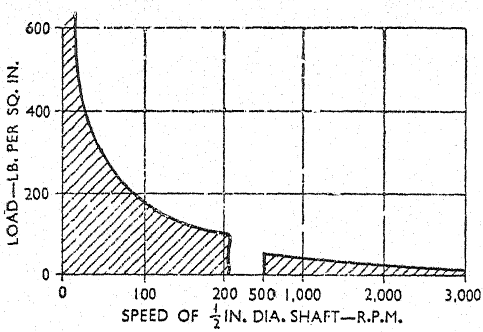


Fig. 7. Range of Application of a Special Self-lubricating Bearing Material of Group 3

Running dry; maximum rise of temperature 65 deg. C.

speed which can be applied if the maximum temperature rise is not to exceed 65 deg. C. The material is manufactured in the form of bushes and split bearings and can easily be adapted to suit other conditions. It safely withstands temperatures up to 120 deg. C. and temporarily even higher ones.

Table 4 illustrates the characteristics of the basic material used as a lubricated non-graphitic bearing.

As outlined above, the basic fibrous material itself is not only used for dry-running, self-lubricating bearings, but is also satisfactory for lighter machinery if water-cooled and, or alternatively, lubricated as explained for the laminated types.

Coming back to the graphitic moulded bearings of felt-like interior structure, it must be said that there was hesitation to use them owing

TABLE 4. COMPARATIVE FRICTION FIGURES OF GROUP 3 BEARING MATERIALS MADE ON A NON-GRAPHITIC LIGNO-CELLULOSE BASE

Lubricant	Brass	Group 3 material
Grease	0·1	0·1–0·15
Oil	0·06	0·045–0·075
Emulsions	0·085	0·07
Water	0·09–0·2	0·02–0·07

to the fear of corrosion of the metal shaft due to atmospheric moisture between the negative graphitic surface and the metal surface. This kind of electrolytic corrosion could be frequently observed when using bearings as briefly described under Group 1. Dispersion of extremely fine graphites, as mentioned above, allows the use of a very much smaller percentage of graphite to obtain satisfactory lubrication. Further, impregnation of the whole bearing with chlorinated materials, such as stable chlorinated hydrocarbons, decreases the water absorption and the electrical conductivity of the bearing body itself. Incorporation of inhibitors, e.g., cobalt chromates dispersed in chlorinated diphenyls, also gives protection against corrosion. Prolonged tests show that practically no corrosion or scoring of the shaft takes place. The bearing surface itself wears very little and continuously polishes the shaft and regenerates its own gliding surface. Up to a certain extent, foreign particles like dust or small abrasives are incorporated into the surface of the bearing, but wherever such contamination is likely to occur, it is advisable to surface-harden the shaft.

Naturally, the wear of this self-lubricating type of bearing material depends upon the conditions of its application. For example, a small split bearing operated under a load of 400 lb. per sq. in. and a speed of 30 ft. per min. showed an average wear of 0·05 mm. for about every 500 hours of continuous running. The material, being homogeneous throughout, can be worn down to a few millimetres' thickness and easily replaced without loss of time, provided the housing is of suitable design.

It should be understood that all plastics used for synthetic resin bonded bearing materials possess an inherent property, generally but sometimes wrongly called "cold flow". In order to prevent deformation due to mechanical strain the housings or the framework have to be designed so as to frame the actual bearing block or lining up to the edge.

Another special design, adapted from a well-known method used for laminated types (Fig. 4), consists of single strips fitted inside a metal housing. As the wear is often higher on the lower parts of the bearings, this arrangement permits of partial replacement.

Another application of the graphitic materials of Group 3 is the use of cones made from these materials in valves and especially in gas valves. The self-lubricating cone, which can be machined and ground easily to close tolerances, fits well into the housing of the valve and no lubricant is required. Valve housings made of cast iron, bronze, or even light metals are only polished but not scored. Prolonged and severe tests have proved that these valves are superior to metal cone types and the coefficient of friction seems to be even lower than recorded with lubricated cast iron cones.

Designers should pay more attention to the possibilities of the bearings discussed under Groups 2 and 3, which can be satisfactorily used for both journal and thrust bearings. They should study the conditions under which these materials are superior to metal bearings and give their attention to proper methods of installation, machining, and cooling. It is necessary, of course, to adapt the design of machinery to the special characteristics of these materials, and this can only be achieved by close co-operation between the plastics industry and the mechanical engineer and designer, to the benefit of industry in general.

Thanks are due to the Verein Deutscher Ingenieure for permission to reproduce Figs. 1, 2, 3, and 5, and to the Morgan Crucible Company, Ltd., for permission to publish data concerning graphitic bearings.

PRINCIPLES OF THE DESIGN OF JOURNAL BEARINGS

By E. Falz*

The high performance of modern sliding bearings was reached only by careful investigation of the nature and peculiarities of lubrication.† In constructing a plain (sliding) bearing for high load capacity and reliability in service, a series of fundamental principles which have been derived largely from theoretical researches on lubrication and partly from practical experience must be considered.

Lubrication in a journal bearing is essentially a mechanical process. Owing to the viscosity of the lubricant and its adhesion to the sliding

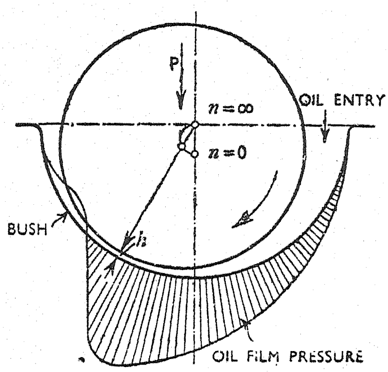


Fig. 1. Oil Film Pressure Diagram of a Journal Bearing with a Smooth Brass

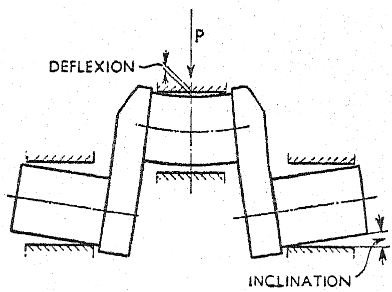


Fig. 2. Inclination and Deflexion of a Laterally Loaded Journal

surfaces, the pressure in the layer of lubricant in the loaded area of the bearing rises when the rotating journal is slightly smaller in diameter than the bore of the bearing. The journal drags, so to speak, the adherent oil into the wedge-shaped clearance and this results in a hydrodynamic lifting of the journal, which assumes a constant dynamic floating position under conditions of perfect lubrication, under which only fluid friction appears (Fig. 1).

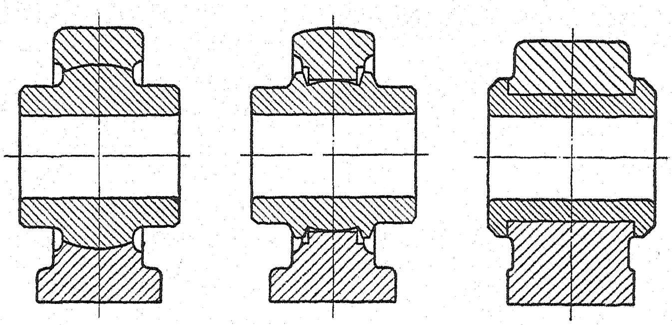
A definite clearance, and therefore a fixed small difference between the radii of brasses and journal is indispensable for obtaining the wedge

* Consulting engineer, Hanover.

† Falz, E. "Grundzüge der Schmiertechnik", Springer, Berlin, 1931.

action of the lubricant and with it the ideal sliding process and lubrication. If the brass is closely fitted to the shaft, there will be no clearance and therefore no fluid friction; mixed friction is set up, accompanied by wear and tear. The greater the clearance, the less will be the load capacity of the bearing, as the layer of lubricant will be forced through more easily. From the point of view of load capacity the bearing clearance must therefore be made as small as other factors will permit. These factors will be discussed later.

Besides ensuring a definite clearance and naturally a truly cylindrical form of the journal and bearing bore, it is necessary to ensure parallelism of the axes of the shaft and bearing. This last requirement is difficult to fulfil on account of the elastic deflexion of the shaft under load. With a laterally loaded shaft (a carrier between two supports),



a *b* *c*

Fig. 3. Types of Housing

a Spherical housing. *b* Tilting housing. *c* Rigid housing.

the elastic deflexion of the shaft becomes on the one hand an inclination (Fig. 2, right) and on the other a bending (Fig. 2, top).

The inclination of the shaft or the central inclination of the shaft in the supporting bearing in respect of the unloaded centre line of the shaft, can only be rectified in practice by a self-aligning arrangement in the casing, because, for example, fitting will not prevent unilateral overloads on bearings under a load which changes in amount or direction, quite apart from the fact that in so doing the requirement that the bearing running surfaces should be truly cylindrical is not met.

The arrangement of the bearing casings themselves comprises two chief types, sliding and tilting. The sliding arrangement is usually spherical (Fig. 3*a*), and this is geometrically free from objection, but very high overloads on the bearing edges are necessary before self-

alignment becomes effective in practice, because for each aligning movement static sliding friction must be overcome, which, as is well known, amounts to $\frac{1}{5}$ to $\frac{1}{7}$ of the shearing force. The tilting arrangement usually takes the form illustrated in Fig. 3*b*. Though theoretically imperfect, this construction is practical, simple, cheap, and, moreover, extremely sensitive, as the self-alignment depends, not on sliding friction, but on rolling tilting. The drawback of the decreased heat transfer between the surfaces of the bearing body and the housing, which touch but slightly, has proved to be hardly noticeable with bearings employed at moderate speeds. This arrangement is also suitable for guide bearings, provided that the axial thrust is not greater than about one-seventh of the lateral load.

If self-alignment is not required and ordinary rigid bearings suffice (Fig. 3*c*), then overloading of the bearing edges will have to be reckoned with, so that the actual load capacity will be far less than that which is technically possible; fitting, as remarked before, will not remedy the rigidity of the bearing, and, further, will spoil the clearance.

The bending of the shaft occurs mainly in the middle of its length when the transverse load comes between two supports, owing to overloading of both bearing edges (Fig. 2, centre). This overloading of the bearing edges while the middle of the bearing does not touch the shaft at all results in a decrease in load capacity and though, fundamentally, it cannot be avoided, it can be moderated quantitatively by diminishing the total deflexion at the middle of the bearing length as much as possible. In principle this can be done in two ways, either by making the shaft thick and rigid or by diminishing the length of the bearing. On highly loaded drives the latter method only is possible because otherwise the construction is not compact enough. With moderate loads, however, short bearings should be used because considerable bending may occur when the supporting bearings are too long.

The load capacity of a bearing depends in essence on the combined action of the bending of the shaft and the end leakage of oil from the bearing. With very short bearings the end leakage is considerable and the harmful effect of shaft deflexion small, whereas with very long bearings the end leakage is small and the harmful effect of deflexion very great. Obviously, therefore, long bearings carry comparatively little load, extremely short bearings not much load, and short bearings very much load. The ratio of the length L to the diameter D should be roughly 1·2 to 0·8. Only where space causes trouble and with light loads may the length be somewhat greater, whereas with maximum loads the L/D ratio may be reduced to 0·5 and even less.

The supply of lubricant directly to the part of the bearing under

active load is impossible because the pressure of the oil wedge in the clearance space is usually incomparably higher than the pressure of the oil itself, as was shown by Beauchamp Tower* (1883–91). It follows therefore that the oil can only be introduced at the unloaded part of the bearing.

The easiest and cheapest method of distributing the lubricant is to provide oil pockets which do not extend quite to the end of the bearing and have a narrow wedge-shaped passage to the bearing surface (Fig. 4, centre). If for economy the wedge-shaped passage between

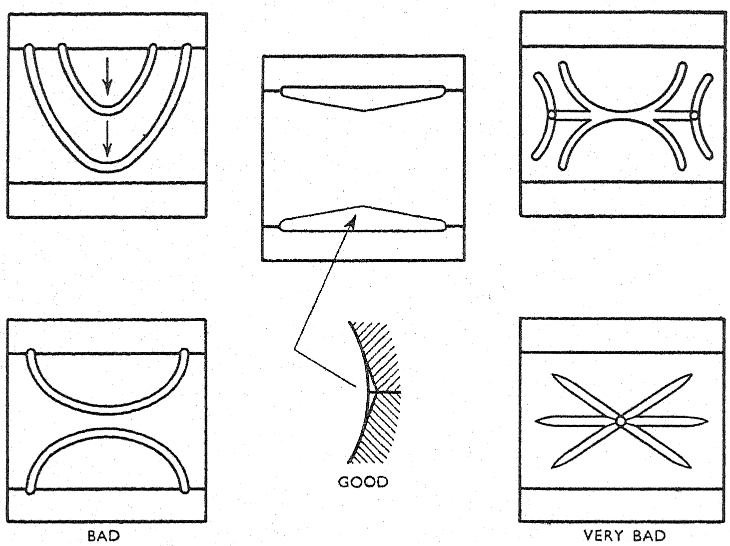


Fig. 4. Four Bad Methods and One Good Method for the Distribution of Oil

the flat pocket and the bearing surface is omitted, the oil at the place of passage will be scraped off and the formation of the oil film will be defective.

The so-called oil grooves which were formerly used (Fig. 4, left and right) not only fail in their purpose but reduce the load capacity and reliability of the bearing. This fact becomes clear, if the diagram of the oil-film pressure for a perfectly smooth half-brass (Fig. 1) is extended to include the change in oil-film pressure for the case in which the bearing has three longitudinal oil grooves going right to the

* Proc. I.Mech.E., 1883, p. 632, etc.

end of the bearing. Fig. 5 shows that in this case the area of the oil-film pressure diagram, and therefore the load capacity, is reduced by some 70 per cent.

The preparation of the bearing surface is of prime importance for

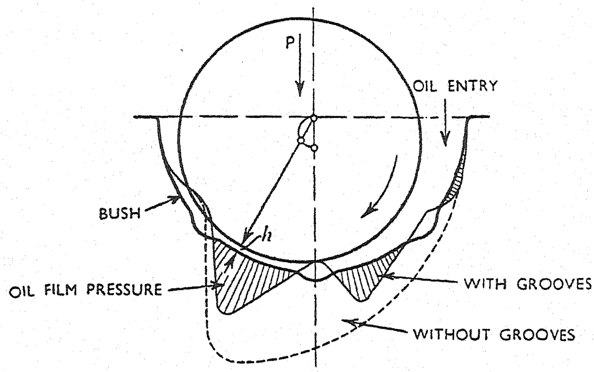


Fig. 5. Oil Film Pressure Diagram for a Bearing with Three Longitudinal Oil Grooves Extending to the End of the Brass

the load capacity of highly loaded bearings. Further, every engineer knows that the surfaces of the journal and bearing bore are not perfectly cylindrical. If one considers, as shown in Fig. 6, the unevenness of the sliding surfaces and that the oil layer between them is often hardly

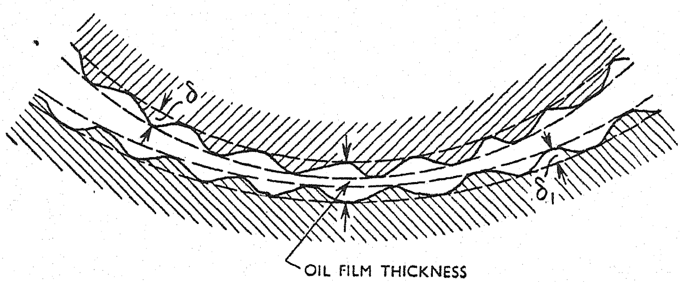


Fig. 6. Diagram of the Uneven Surfaces of Brass and Journal

greater than 0·01 mm. in thickness, it becomes obvious that the accuracy of the cylinder and the surface evenness of the sliding surfaces must satisfy severe requirements, particularly with mixed friction, if reliability and the least possible wear are to be attained. Thus if the unevenness of the sliding surfaces is reduced by one-half (Fig. 7),

then the minimum admissible thickness of the oil layer is also halved. This actually means that the load capacity is doubled, i.e., according to circumstances, the conversion of mixed friction and wear into fluid friction and freedom from wear. As this doubling of load capacity by reducing the unevenness of the sliding surfaces by one-half is quite feasible in modern practice, the requirement for the highest possible finish of sliding bearing surfaces is not a superfluous luxury but a most important means of increasing load capacity, reliability, and service life.

The improvement of the running surface of the journal can be accomplished in the usual way by lapping with fine emery powder and

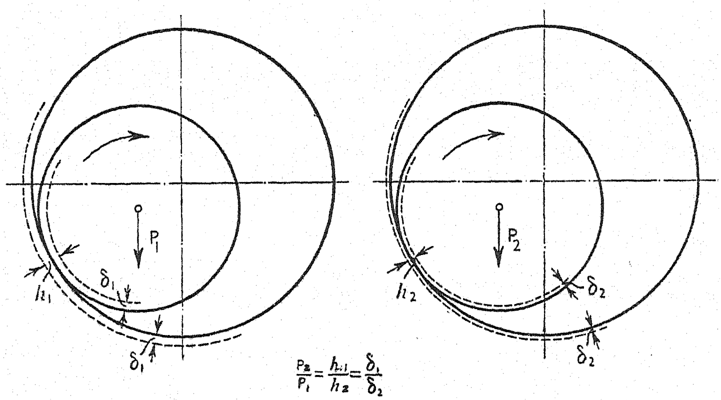


Fig. 7. Illustration of the Doubled Load Capacity Obtained by Halving the Unevenness of the Sliding Surfaces

paraffin oil, and with small shafts this is an easy matter. Shafts that have been ground act mainly as milling tools and therefore are causes of wear. The fine machining of bearing running surfaces is best carried out—according to the available equipment—with diamond or hard-metal tools on a lathe running without vibration or on a special high-speed drilling machine.

There is little advantage in combining an unhardened steel journal with brasses made of tin alloy or copper bronze, as the shaft would inevitably wear, so unhardened steel journals should only be used with whitemetal or lead-bronze linings. Tin alloy, and particularly phosphor-bronze, should only be used on hardened shafts which have been ground and polished.

As regards the choice of clearance, this must be proportionately small in order to obtain the maximum load capacity, and though this

statement is correct, it should not lead to clearances which are too small. In determining the practical clearance, the deflexion of the shaft, the speed, and the bearing temperature have to be considered. In compact constructions the bending of the shaft is usually very small, so that it can be neglected in computing the clearance. The influence of the speed of revolution on the clearance is felt only at high speed, so that the question will not be discussed here. Details are given elsewhere.*

For the choice of correct dimensions for the clearance, the bearing

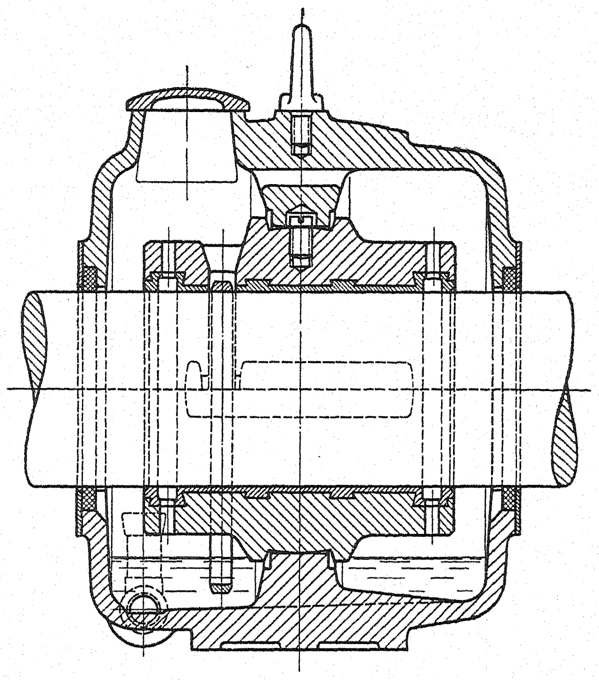


Fig. 8. Falz Oil Ring Bearing

temperature is important if the bearing runs at comparatively high temperatures. In that case a distinction should be made between the cold clearance and that which results when the bearing runs warm. If a comparatively small clearance has been chosen to ensure adequate load capacity, the warm clearance must be sufficiently greater than the cold clearance to allow for the effect of expansion. Space is not

* *Schweizerische Bauzeitung*, 1934, "Lagerspiele für hohe Drehzahlen".

available to deal with the effect of expansion in numerical detail. Papers on this subject have been published by the author.*

The guiding principles for the construction of plain bearings which have been discussed above are embodied in the practical application illustrated in Fig. 8, representing a simple machine bearing. The body of the bearing is supported on a narrow cylindrical casing, and as it is self-aligning, displacements of the shaft are automatically corrected. Oil is admitted through a roomy oil hole with a spring cover, and a loose oil ring supplies abundant lubricant under any direction of loading. Excess oil from the shaft runs into circular grooves in the body of the bearing and is conveyed to the bottom of the casing, felt rings being used to prevent the entry of dust into the bearing.

**Petroleum*, 1933, vol. 29, September Supplement, p. 2, "Der Wärmeausgleich im Gleitlager"; *Petroleum*, 1934, vol. 30, p. 2, "Das Lagerspiel bei höheren Temperaturen".

SOME EXPERIMENTS WITH WATER-LUBRICATED RUBBER BEARINGS*

By A. Fogg, M.Sc.,† and S. A. Hunwicks, B.Sc. (Eng.)†

For some time rubber has been used as a bearing material where it is either impracticable or inconvenient to lubricate with oils or greases, such as stern-tube bearings of ships, water pump bearings, etc. When lubricated with water, rubber bearings have apparently worked satisfactorily, both for length of service and low friction, whilst the softness of the rubber has allowed particles of sand and grit to travel through the bearing without undue scoring of the shaft.

The present investigation has been carried out in order to obtain data on the load-carrying capacity and friction losses of water-lubricated rubber bearings at various speeds; comparison has also been made with a water-lubricated bronze bearing.

The Bearings Used. The performance of two types of rubber bearing has been determined: (1) a smooth cylindrical bearing without grooves of any kind, and (2) a fluted bearing having eight longitudinal grooves, thus giving contact with the shaft along a series of strips of rubber running the full length of the bearing. The latter is the type generally in use. The bushes consisted of steel shells lined with rubber about $\frac{1}{4}$ inch thick. The fluted rubber bushes were supplied as being suitable for a 2·000-inch diameter shaft. The plain rubber bush was ground to the size required in the Engineering Department workshop of the National Physical Laboratory. The bronze bush used for comparison was finished to a very fine surface by polishing with the finest grades of emery paper. All the bushes were $2\frac{1}{4}$ inches long.

Experiments. The experiments were made on a National Physical Laboratory journal bearing machine.‡ The coefficient of friction was measured under various loads up to 250 lb. per sq. in. of projected area and over a range of speed up to 1,800 r.p.m. Preliminary observations were made on a fluted rubber bush to determine the best procedure for covering the range of load and speed. This bush was damaged by running under conditions of high load and low speed—

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* Work carried out for the Lubrication Research Committee of the Department of Scientific and Industrial Research.

† Engineering Department, National Physical Laboratory.

‡ See Group I, p. 138. "The Performance of Complete Clearance Bearings as Affected by Changes in Load, Speed, Clearance, and Lubricant", by C. Jakeman and A. Fogg.

outside the range of fluid film conditions—and in subsequent tests all the observations in the region of fluid film lubrication were completed before testing the region of partial boundary and fluid film lubrication. Tests were made at constant load, starting at the highest speed and reducing speed until the friction showed signs of increasing. The load was then increased to the next value and similar observations made. This was continued up to about 100 lb. per sq. in.—the bush still showing no signs of wear—and before increasing the load beyond this value, the effect of further reducing the speed within this load range was investigated. Care was taken to ensure that the bush did not wear appreciably by not decreasing the speed too far, so that further observations could be made on the same bush at loads above 100 lb. per sq. in. before there was any appreciable change in the shape of the bush. The machine was run under each load and speed until the temperature of the bearing was steady, the coefficient of friction then being measured.

Observations were made on the following bushes (the rubber bush measurements are correct to within $\pm 0{\cdot}0005$ inch) :—

(1) Plain cylindrical rubber bush having a diametral clearance of 0·003 inch.
(2) Fluted rubber bush ground out to give a diameter across opposite faces equal to the shaft diameter. When finished it had a diametral clearance of 0·001 inch. (This bush was originally 0·002 inch *less* than the shaft diameter and, as the machine could only be started with difficulty, the bush was ground out as stated above, after making a few observations.)
(3) Fluted rubber bush ground out to give the same clearance as the plain cylindrical bush; measured clearance 0·0025 inch.
(4) Bronze bush having a diametral clearance of 0·0022 inch.

The water was pumped to the bearing, the supply being sufficient for water to flow out from that part of the bearing under load. Measurements of the fluid pressure in the film were made in order to verify that a pressure film was present in the bearing and, therefore, that the supply of water was sufficient for fully lubricated conditions.

Preliminary Experiments on Fluted Rubber Bush. Observations of the friction were made with the bush running (1) with the line of loading passing between two facets, and (2) with the line of loading passing through the centre of one facet. With a load of 11·1 lb. per sq. in. there was little difference between the two positions; at 22·2 lb. per sq. in., however, the friction in the second position was about 60 per cent of that in the first position over the whole speed range. All

subsequent tests on the fluted bushes were, therefore, made with the line of loading passing through the centre of one facet.

Plain Cylindrical Rubber Bush. Measurements of the friction were made up to a load of 250 lb. per sq. in., at which load the first signs of wear became apparent. The bush had been lightly marked over an arc extending nearly 180 deg. in the early stages, but there was no measurable wear until the load reached 250 lb. per sq. in. Curves showing coefficient of friction plotted against speed for loads from 11·1 to 250 lb. per sq. in. are reproduced in Fig. 1. The coefficient of friction falls as speed decreases until a certain speed is reached, after which it increases at an increasing rate. The speed at which the minimum value of friction is reached increases as the load increases, being about 300 r.p.m. at 11·1 lb. per sq. in. and about 1,500 r.p.m. at 200 lb. per sq. in.; at 250 lb. per sq. in. the minimum value is at or beyond the upper limit of the speed range investigated. At speeds above that at which the minimum value of friction occurs, the bearing is considered to be operating under complete fluid film conditions and below this speed, under partial boundary and fluid film conditions. The fact that the transition point occurs at higher speeds as the load increases indicates that the bearing is behaving in a similar manner to a normal oil-lubricated metal bearing. The values of ZN/P (Z=viscosity of water in centipoises; N=speed of journal in revolutions per minute; P=load on bearing in pounds per square inch) at the transition point vary, however, from about 25 at the lowest load to about 7 at the highest load, but this is probably due to the flexibility of the rubber giving an effective clearance varying with load. Although at the higher loads the value of the coefficient of friction does not rise rapidly at speeds below that at which minimum friction occurs, it would be unsafe to run for extended periods in this region as rapid wear would probably occur.

Fluted Rubber Bush (*Clearance:* 0·001 *inch*). Measurements of the friction were made up to a load of 250 lb. per sq. in. (Fig. 2). From 150 lb. per sq. in. upwards, there was a steady rise in friction as the speed was reduced, indicating partial fluid film conditions only throughout the speed range, but there was no appreciable wear until the load reached 250 lb. per sq. in. The friction-speed curves obtained are similar to those for the cylindrical rubber bush in that the friction increases at an increasing rate as the speed decreases after a certain speed has been reached. An interesting feature of these curves, however, is the constancy of friction over the range of speed from the maximum down to that at which the friction begins to rise. If the

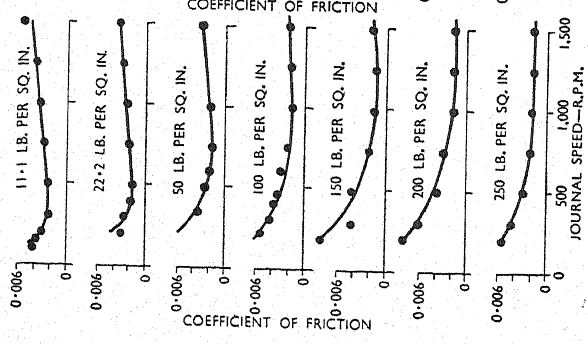

Fig. 1.
Plain Rubber Bush
Clearance 0·003 inch.

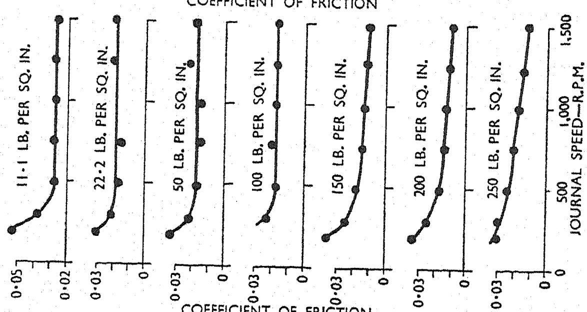

Fig. 2.
Fluted Rubber Bush
Clearance 0·001 inch.

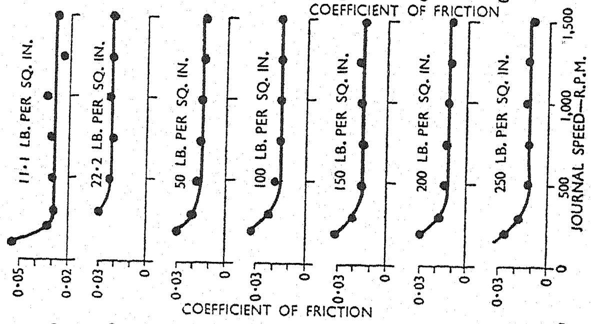

Fig. 3.
Fluted Rubber Bush
Clearance 0·0025 inch.

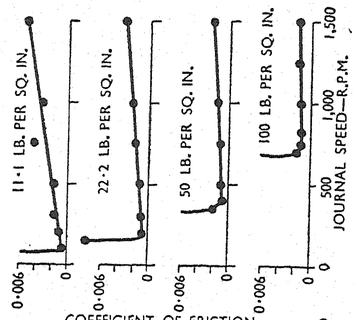

Fig. 4.
Bronze Bush
Clearance 0·0022 inch.

bearing is running under complete fluid film conditions over this range of constant friction, this feature could be explained by a clearance varying with speed, which, however, is not easy to visualize. The actual value of the coefficient of friction with this bush is many times greater than with the plain rubber bush (the minimum value in this case being about 0·01 as against 0·001–0·002 with the plain bush) and this shows that the mechanism is different in the two cases. The difference in friction in the two cases under any one set of conditions might be attributed to turbulence in the bearing grooves, although the constancy of friction with speed with the grooved bush cannot be explained on this basis.

Fluted Rubber Bush. (*Clearance:* 0·0025 *inch*). The second fluted bush was given the same clearance as the plain bush to determine whether the difference in friction was due to the different clearances. The results (Fig. 3) are very similar to those for the fluted bush, clearance 0·001 inch, the only appreciable difference being at 200 and 250 lb. per sq. in., constant friction over the speed range 500–1,500 r.p.m. being maintained in this case up to the highest load tested. With both fluted bushes the speed at which the friction begins to rise is almost independent of load. The values of ZN/P at the lowest values of friction vary from about 25 at the lowest load to about 10 at the highest.

Bronze Bush. As before, all measurements of friction in the complete fluid film region were completed before the region of partial film lubrication was explored. The load was not increased above 100 lb. per sq. in. as, at this load, the bearing would not run at speeds below about 700 r.p.m. The lowest coefficient of friction recorded is just below 0·001, and the coefficient of friction–speed curves (Fig. 4) are very similar to those for the plain rubber bush from the highest speed to that at which the friction begins to rise. The transition from fluid film to boundary conditions is, however, much more severe with the bronze bush than with any of the rubber bearings, the friction rising very rapidly once the minimum is passed. This feature is of extreme importance as, unless conditions are such that a complete fluid film is formed, the bronze bush will either give extremely high friction or seize. The value of ZN/P at the lowest value of friction is about 7 for all loads; this is of the same order as for an oil-lubricated bronze bearing before running-in.

Conclusions. Rubber bearings of the type used in these experiments work satisfactorily when lubricated with water up to loads of about

200 lb. per sq. in. and are most efficient at speeds above about 500 r.p.m. At speeds above this figure, the coefficient of friction of fluted rubber bearings is almost independent of speed and load and that of the plain cylindrical rubber bearing increases slightly with speed. The value of the coefficient of friction, however, is considerably lower under all conditions tested for the plain bush than for the fluted bush, which is the type generally in use. The superiority of the plain bush in this respect might be, to some extent, nullified where the water contains particles of grit and sand, as such particles would probably escape more easily from the fluted bush and prevent scoring. This aspect has not been investigated up to the present. The fluted rubber bush has a smaller friction loss when the line of loading passes through the centre of one facet than when it passes through the centre of a groove. An increase in the diametral clearance from 0·001 to 0·0025 inch gives easier starting, but does not appreciably affect the friction when running.

At high speeds, the coefficient of friction of the bronze bush is of the same order of value as that of the plain rubber bush and much lower than that of the fluted bushes. When the speed is reduced, however, so that complete fluid film conditions no longer exist, the rise in friction with the rubber bearings is gradual and, provided the load is not excessive, no appreciable wear takes place. The bronze bearing on the other hand, seizes at the higher loads and, at lower loads, the friction loss is excessive and rapid wear occurs.

PIVOTED PAD BEARINGS

By J. Hamilton Gibson*

Thirty years ago oil-lubricated bearings for all classes of machinery were much the same as they had been since the middle of last century. Complete journals comprised solid top and bottom brasses, thrust bearings were of the multi-collar type with elaborate means for individual and collective adjustment, and footstep bearings were of the crudest form. Latterly, forced lubrication methods had improved matters as regards excessive wear and engineers had come to realize by long experience that the lubricant should be introduced at the point of minimum bearing pressure, also that a good deal could be done by a logical arrangement of oil grooves.

Textbooks of the period, illustrating bearings for crankshafts, crankpins, crossheads, and guides, also thrust collars, indicate solid half-brasses or flat sliding surfaces rigidly secured in their respective housings and generally faced with anti-friction whitemetal. Later examples showed an appreciation for the right disposition of grooves, i.e. wide, deep, well-chamfered channels cut square across the bearing surface, which was thus divided into a series of rectangular panels, each getting its quota of lubricant right across its leading edge. In some cases, especially where forced lubrication was employed, the grooves were "stopped" at the ends to prevent undue escape of oil.

Beauchamp Tower (1883) found (Fig. 1) that a shaft running in an oil bath with a loaded half-brass resting on top not only carried the oil up and under the brass, but automatically created a considerable pressure in the oil film, which, of course, varied in thickness from the "on" to the "off" side due to the relative shift of shaft and brass. These early experiments led Osborne Reynolds to formulate a mathematical theory of lubrication, and inspired others like Professor Goodman and Sir Thomas Stanton to follow up with further experiments, all of which corroborated Tower's results. Journal bearings were designed to take advantage of the principles enunciated and, as is well known, many of these ran for long periods with very little friction and no appreciable wear, although it was recognized that this effect was largely fortuitous.

The Thrust Bearing

Meanwhile the problem of the thrust bearing was becoming increasingly acute as the speeds and powers of ships grew greater, to

* Michell Bearings, Ltd.

say nothing of developments in land power stations where big thrust loads were encountered. To meet these increasing axial loads engineers had recourse to the obvious method of increasing the bearing surface by adding more and more thrust collars, and in extreme cases even duplicating the thrust block. It is unnecessary here to recount the troubles that were then experienced, except to point out that the inevitable differential expansion between the thrust shaft and the thrust block upset the nice adjustments made when both were at the same temperature, and the load thus became more and more con-

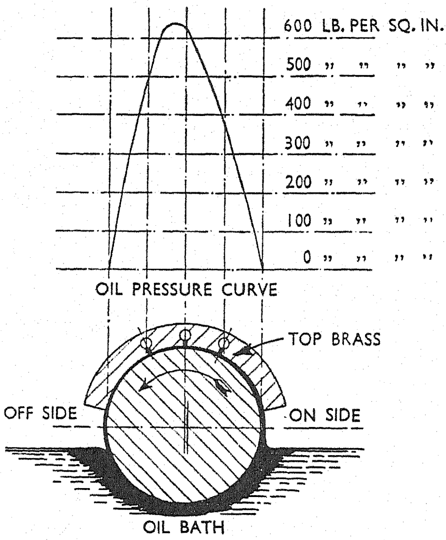

Fig. 1. The Original Experiment of Beauchamp Tower

centrated on one or two collars at one end, with dire results. It was considered undesirable to design such multi-collar thrust bearings for loads over 50 lb. per sq. in., and good merchant ship practice aimed at considerably less.

During the years 1902–4 A. G. M. Michell (1905) made a study of Reynolds's theory with a view to applying the principle of tapered film lubrication as found in journal bearings to the problem of the thrust block. He carried out a series of experiments to ascertain whether the required condition could be provided in a thrust bearing, namely, the maintenance of the rubbing surfaces at such relative positions to each other as would allow the formation of wedge-shaped films of oil

between them. This involved the employment of pivoted bearing pads and, having satisfied himself that the requirements had been fulfilled, Michell took out a provisional patent (No. 875 of 1905) entitled "Improvements in Thrust and Like Bearings", remarkable as much for its brevity as for the importance of its subject matter. The salient features are worth quoting:—

"According to this invention only one of the bearing parts has the form of a continuous collar, ring or disc. This may be either the fixed or the moving part. The other bearing part, in place of being also a continuous ring or disc, consists of two or more separate plates or blocks, each forming substantially a sector of such ring or disc.

"Each block is pivoted at a point somewhat behind its centre of figure, the portion of the total bearing pressure which the block carries being transmitted to it by such pivot.

"The object of this arrangement is that the film of oil between each block and the surface on which it bears may be more compressed or restricted at the rear end than at the front or leading end of the block, a condition which favours the entry of oil between the surfaces at the leading end."

Reference to Michell's complete specification shows that the above-mentioned pivoted sectors, now long known as "thrust pads" (Fig. 2), were even then correctly proportioned, that is to say, the face of each one was approximately square and capable of pivoting or rocking on a radial line behind and beyond its centre of figure in relation to the direction of shaft rotation. In some subsequent designs these tilting pads were made kidney-shaped, i.e. the circumferential length was very much greater than the radial width and the load was concentrated on a rounded point at the centre of figure. Experience having proved that these departures were wrong, practice long ago reverted to the original form. The thrust pads are now solid segments of a flat ring, the mean circumferential length of each being preferably rather less than the radial width, and they are merely "stepped" at the back so as to rock freely on a radial line (Fig. 3), suitable stops preventing rotation with the shaft. With these general proportions and an adequate supply of suitable oil a single-collar thrust bearing can carry enormous loads. Thus Michell thrust bearings have been tested to a pressure of 12,000 lb. per sq. in. without failure of the oil film (Broom 1914).

Normal working conditions, however, rarely call for more than 500 lb. per sq. in., and 250 lb. is quite usual. If the shaft is required to start rotation under load it is well not to exceed 350 lb. Tilting pads may be allowed to rock about their centre of figure to avoid "handing"

where rotation under moderate load is in either direction. The single-collar thrust bearing with tilting pads is admirably adapted to marine propulsion. The highest attainable horse-powers are well within its capacity and offset pads of the same hand are suitable for driving ahead or astern. Generally, plain bath-lubrication suffices, but where cooled circulated oil is available, as in turbine installations, it is, of course, utilized. Pivoted pad thrust bearings work equally well on vertical shafts, but as these usually start under loaded conditions they are not

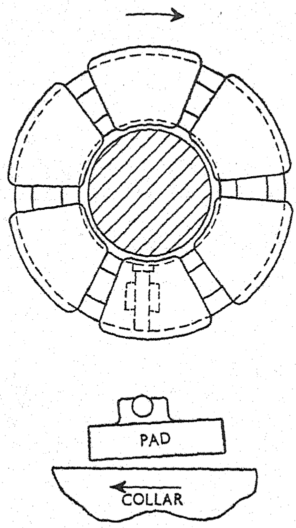

Fig. 2. Michell Thrust Pad

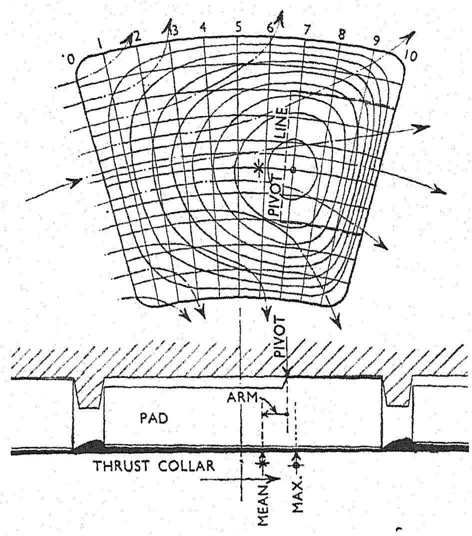

Fig. 3. Oil Flow and Pressure Lines on Pivoted Pad

such an easy matter as marine propeller thrust bearings, which make several revolutions before there is any appreciable load.

A remarkable thing about the pivoted-pad thrust bearing is that it took so long for its outstanding merits to be recognized. Indeed it may be said that it was the advent of the marine geared turbine that forced the pivoted pad thrust bearing on the notice of the engineering world; and soon thereafter it became universal practice.

Pivoted Pad Journal Bearings

About the period 1911–12, Michell patented pivoted pads for journal bearings, realizing that the functioning of the best "solid-brass" bearings was largely fortuitous, and that positively good results were

only attainable by the employment of tapered oil films all round the shaft instead of at one or, at most, two points of the circumference, as in ordinary journal bearings.

The author was one of the first to adopt the Michell thrust bearing in shipbuilding practice, and being impressed with its novelty and mechanical perfection decided to experiment with pivoted pads in a journal bearing (Gibson 1917). Stated briefly, the rig (Fig. 4) consisted of a housing split horizontally and capable of being closed by a weighted lever giving a powerful nut-cracker effect on the shaft which was of 12 inches diameter. Five centre-pivoted pads only 2 inches long axially were fitted in each half of the housing. At 1,320 r.p.m. and a projected mean bearing pressure of 900 lb. per sq. in., the circulated oil

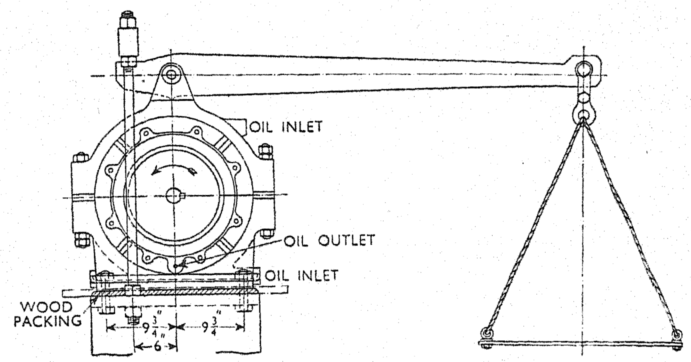

Fig. 4. "Nut-cracker" Test Rig for Michell Journal Bearing

outlet temperature never rose above 117 deg. F., which represented a rise of 41 deg. The coefficient of friction was only 0·0034, and after prolonged tests the pads showed no signs of wear. Further tests were made with three pivoted pads in the top and bottom halves of the housing. These were 3 inches long axially, i.e. one-quarter of the shaft diameter, and small holes were drilled in each pad to ascertain the oil film pressures generated. Steady pressures were recorded ranging from 3,000 to 3,200 lb. per sq. in. at the top and bottom pads respectively, and under the four side pads 1,100 to 1,800 lb. per sq. in. Again, no wear could be detected after prolonged runs at these loads. The Admiralty decided to try pivoted pads in the turbine bearings of a 40,000 s.h.p. twin-screw geared turbine destroyer which was then building.* The low-pressure turbine had spindles of 9 inches diameter, running at 1,800 r.p.m. at full power. The length of the bearing

* See Proc. I.Mech.E., 1929, p. 875; 1932, p. 525.

surfaces in the ordinary bearings was $10\frac{1}{2}$ inches as against only $3\frac{3}{4}$ inches in the pivoted pad journals, and the corresponding loads were respectively 120 lb. and 550 lb. per sq. in. The shorter bearings ran consistently 18 deg. F. cooler than the longer bearings, and after 12,000 miles sea service the pivoted pads were found to be in perfect condition, while it was reported six years later that there was no perceptible wear. Pivoted-pad journal bearings (Fig. 5) were fitted in several high-powered destroyers for the British and foreign navies, not only for the turbines but also throughout the gear cases, and so far as is known no replacements or even remetalling of pads have ever been required.

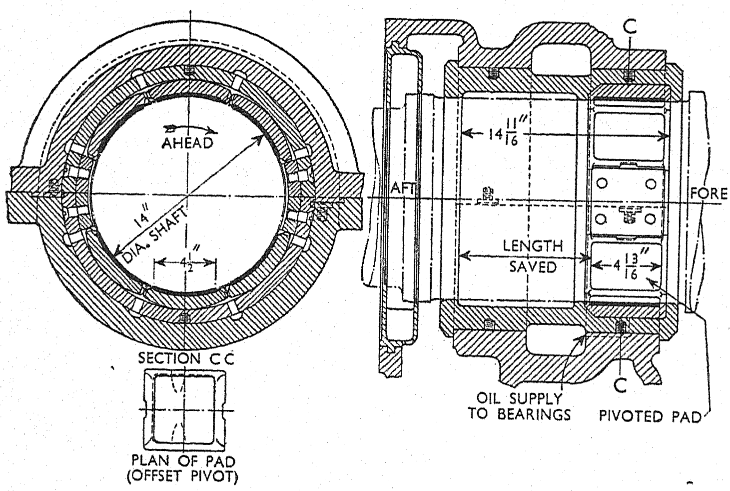

Fig. 5. Conversion of Main Gearwheel Bearing in the S.S. *Gouverneur Général Chanzy*, showing Saving of Space by Using Pivoted Pads

Shaft, 14 inches diameter; speed, 140 r.p.m.; bearing pressure, 370 lb. per sq. in.

Pivoted pad journals adapt themselves neatly to the steadiment bearings usually associated with vertical spindle pivoted-pad thrust bearings now extensively used in hydro-electric power and pumping stations (Fig. 6). They have also been widely adopted in place of ordinary horizontal journal bearings for shafts and Pelton wheels and for isolated transformers in out-of-the-way mountainous regions. Such self-contained, wear-free journal bearings are particularly valuable for tunnel shafting, as such shafting is sometimes difficult of access and the maintenance of alignment is always of considerable importance. Pivoted-pad tunnel bearings are now fitted in many ocean liners,

including vessels like the *Empress of Britain* (Fig. 7) and the *Queen Mary*. In naval vessels the shaft tunnel arrangements are different and there are fewer bearings. The space available is often very limited,

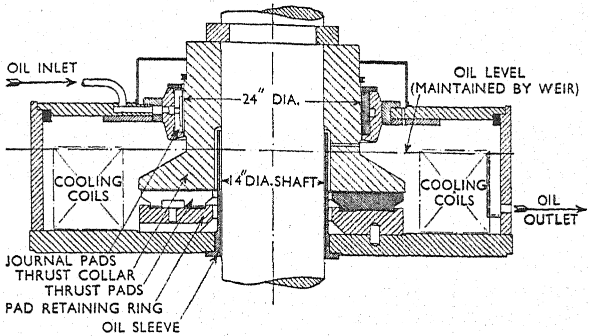

Fig. 6. Vertical Pivoted-Pad Thrust Bearing

and short compact bearings which will run with the minimum of attention for prolonged periods, preferably without water cooling, are

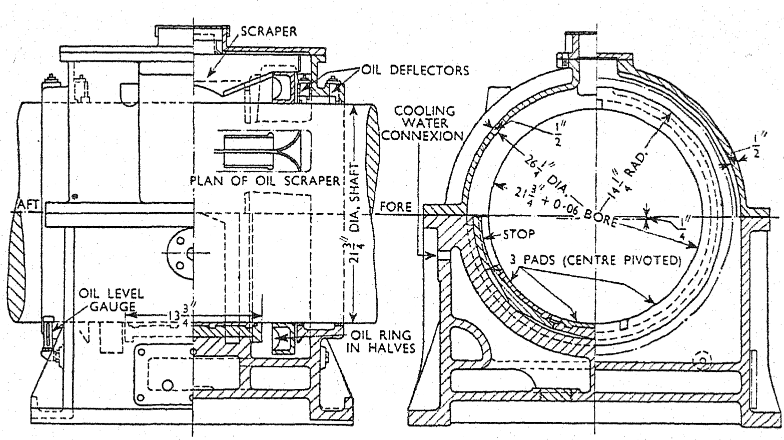

Fig. 7. Pivoted-Pad Self-lubricating Bearing for Tunnel Shafts in the Quadruple-Screw 60,000 s.h.p. Liner *Empress of Britain*

necessary. The pivoted-pad journal bearing fulfils every such requirement and is now standard practice in all classes of warship.

In reciprocating machinery there is a tendency for the pivoting ridge to indent the housing and therefore pivoted-pad journal bearings are not

recommended, though prolonged tests show that the bearing surfaces function perfectly. A locomotive ran 80,000 miles on six 10-inch coupled-axle bearings, and on an Air Ministry test of a 3-inch by 3-inch big-end bearing at 3,000 r.p.m. a load of 3,000 lb. per sq. in. was carried for a 30-hour continuous run. The rubbing surfaces were not affected and it is interesting to note that in the latter case the six pads rocked on the crankpin and rubbed on the inside bore of the big end, thus taking advantage of centrifugal retention of oil on these surfaces.

The principle of tapered-film lubrication is now fully established. Its practical application depends on the correct design, construction, and operation of the complete bearing, whether thrust or journal or both combined. But no working bearing is complete without oil and it is of the utmost importance that an ample supply of clean oil should circulate constantly through and between the loaded parts. Bearing design requires as its chief aim the coaxing (not forcing) of the main oil streams into and out of the specially prepared oil clearance spaces between plane or cylindrical rubbing surfaces. Pivoted bearing-pads will do the rest automatically and perfectly, if left free to accommodate themselves to the varying conditions of speed, load, and viscosity.

Generally considered, and as pointed out in the opening paragraphs of this paper, the Reynolds theory of lubrication operates in more or less degree in every journal bearing. Its maximum effect, however, is experienced only in pivoted pad journals of the type described, and as regards the application of the same theory to flat bearing surfaces, e.g. thrust blocks, Michell (Bergstrom 1920) was undoubtedly the first to realize the possibilities and to show how it could be accomplished.

References

Bergstrom, E. M. 1920 Proc. I.Mech.E., p. 55.

Broom, H. S., on Newbigin's paper. 1913–14 Proc. Inst. C.E., vol. 196, p. 255.

Gibson, J. H. 1916–17 Trans. Liverpool Eng. Soc., vol. 38, p. 330.

Michell, A. G. M. 1905 *Zeitschrift für Mathematik und Physik*, vol. 82.

Tower, B. 1883 Proc. I.Mech.E., p. 632.

STEAM TURBINE JOURNAL BEARINGS

By H. L. Guy, F.R.S., M.I.Mech.E. (*Member of Council*),*
and D. M. Smith, D.Sc., A.M.I.Mech.E.†

The authors have collected a statement of current turbine journal bearing practice through the co-operation of the six principal makers, who collectively produce over 90 per cent of the land turbines built in this country. Table 1 summarizes certain aspects of the current practice of these manufacturers.

Bearing pressures vary normally between 57 and 150 lb. per sq. in. of projected area, but in particular cases are as low as 27 lb. per sq. in. or as high as 200 lb. Surface speeds of journals up to 157 ft. per sec. are commonly used, while 177 ft. per sec. represents the maximum in current practice. While these pressures and speeds represent normal practice, at least one of the makers has in a special application operated pinion bearings in a turbine-driven set at pressures up to 285 lb. per sq. in. and surface speeds up to 232 ft. per sec.

In general, turbine bearings are now very much shorter in proportion to their length than formerly. A ratio of length to diameter of 1 to 1·25 represents normal practice, with a minimum in special cases of 0·75. In contrast the ratio is often as high as 2 for turbo-generator bearings.

One maker finishes the bore with temporary liners in the horizontal joint so that the horizontal clearance is larger than the vertical clearance. The other makers employ circular bore bearings, with the diametral clearance varying from 0·001 inch per inch to 0·003 inch per inch, the average of present practice being 0·002 inch per inch. For very small bearings, the clearance would not be proportional to the diameter and a minimum diametral clearance of from 0·003 inch to 0·005 inch is observed. With a clearance of over 0·003 inch per inch vibration is experienced, while if the clearance is below 0·001 inch per inch the bearings may become unduly hot.

Five out of the six makers reduce the extent of initial bedding of bearings in their shops to a minimum. During service most of the makers permit very little bedding and impose an upper limit of the arc showing bedding of about 30–40 deg. on each side of the vertical.

Four makers usually support their bearings on detachable pads which can be used to vary the position of the journal relative to the

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TABLE 1. TURBINE JOURNAL BEARING PRACTICE

Manufacturer	A	B	C	D	E	F
Load on projected area, lb. per sq. in.						
Minimum	50	45	30	27	30	—
Normal	—	100 to 150	100	80 to 130	70 to 120	57 to 114
Maximum (practice)	146	160	130	170	150	200
Surface speed, ft. per sec. Normal	150	Up to 150	130	Up to 157	Up to 144	—
Maximum (practice)	170·5	177	160	170	157	165
Ratio $\frac{\text{length}}{\text{diameter}}$ Minimum	0·9	0·8	1	0·75	1	1
Normal	1 to 1·5	1	1·25	1 to 1·5	1·25	1·6
Maximum	2·5	1·5	2·0	2	1·5	—
Clearance per inch diameter . inches	0·0015 to 0·002	0·002	0·0015	0·002 to 0·003	0·001 to 0·002	0·001 Vertical 0·004 Horiz.
Minimum thickness of whitemetal; journal dia. 4 inches . inches	0·16	$\frac{3}{16}$	$\frac{3}{16}$	$\frac{1}{4}$	$\frac{5}{16}$	0·2
Minimum thickness of whitemetal; journal dia. 12 inches . inches	0·28	$\frac{3}{8}$	$\frac{1}{4}$	$\frac{3}{8}$	$\frac{3}{8}$	0·5
Percentage composition of whitemetal:—						
Tin	76	83·4	82·9	86 72–76	77	86 78
Antimony	10	8·3	8·3	8·5 6–7	14	8·5 13
Copper	6	8·3	8·3	5·5 4–5	9	5·5 6
Lead	8		0·5	13·5–16·5		3
Approximate oil quantity, gal. per min.:—						
10×10-inch bearing, 3,000 r.p.m.	16	26·7	11·6	11·3	13·7 to 20·6	—
20×20-inch bearing, 1,500 r.p.m.	48	75	46·4	45·2	54 to 82	—

bearing housing. Two makers invariably support their bearings in solid cylindrical seats, as do the other makers in special instances. Spherical seats for the support of bearings are adopted by three of the makers for combined thrust and journal bearings and by two of the makers for large journal bearings.

Compositions used for whitemetal linings are shown in Table 1. Two makers use two different mixtures, that with the higher tin content being employed for higher temperature work. Two makers extensively use centrifugal casting to form the whitemetal shells; the others cast their linings stationary. All the makers vary the minimum thickness of whitemetal with the journal diameter, the figures for 4-inch and 12-inch diameters being shown in Table 1. Various methods are used for locking the whitemetal lining into the bearing shell, which is almost invariably made of either cast iron or steel. In typical examples three makers use closely pitched circumferential dovetail grooves with few axial grooves. Two other makers use six wide axial grooves with very few circumferential grooves, and one maker uses several circumferential grooves with two axial grooves on the vertical centre line and two axial grooves adjacent to the horizontal centre line.

Two makers introduce oil through long slots on both sides in the horizontal plane. One maker introduces the oil on the falling side with a deep narrow channel cut round the top half bearing. Another introduces oil at the rising side to a deep circumferential channel in the top half from which small channels run axially outward. A fifth maker, for special high-speed bearings, introduces oil for lubrication through a wide, long recess on the falling side of the journal and below the horizontal line, while additional cooling oil is admitted through a long narrow slot at the horizontal diameter on the rising side with a special drain passage on the opposite side to draw off this cooling oil. The sixth maker, for high-speed bearings, uses a scraper type bearing in which oil is introduced through a long wide passage on the rising side and above a moveable scraper which diverts the oil rising from the bottom half of the journal into drain passages provided along the whole length.

For the assessment of the oil quantity with which the bearing is to be supplied and for the design of the lubricating system, the various makers use approximate formulæ which give results for specific bearings shown in Table 1. For a bearing 10 inches in diameter and 10 inches long, running at 3,000 r.p.m., different assessments range from 11·3 to 26·7 gal. per min., while for a bearing 20 inches in diameter and 20 inches long, running at 1,500 r.p.m. the assessments range from 45 to 82 gal. per min.

Most of the makers have, in certain instances, adopted special

designs of bearings to deal with peculiar vibrations. One reports satisfactory use of Newkirk stabilized bearings to deal with oil film vibration. A second has used special grooves in the bottom half of the bearing to break up the oil film on turbines which run through their critical speed. Two other makers have used concentric multi-bush type bearings on high-speed machines, but one of these reports that his experience with these has not been very satisfactory and that he has used spring-loaded " slip " type bearings to deal with such cases, but again with not very satisfactory results.

Tilting pad journal bearings have been used satisfactorily by one maker for high-speed machines in place of ball bearings. Tilting pad journal bearings have been used experimentally by two other makers, but neither has found a place for this type of bearing in normal work, objection being raised on account of the difficulty of controlling oil supply and the loss of assistance in damping out vibration given by a rigid bearing of moderate length.

Five of the makers fit high-pressure lifting devices on large journal bearings when continuous barring apparatus is employed. The high-pressure oil is supplied through special passages near the vertical centre line of the bottom half where it is admitted to the space between the journal and the bearing through several shallow recesses. The oil is usually supplied by a reciprocating oil pump which has a separate pump barrel supplying each bearing equipped for high-pressure oil. One maker uses a single gear type pump which supplies all the bearings equipped for pressure lifting in parallel. The pressure specified for the oil pump varies in each maker's practice from 1,000 to 2,000 lb. per sq. in. The actual pressure employed is determined by the rotors and bearings themselves and is usually less than 1,000 lb. per sq. in.

The increasing requirements of modern and future machines have revealed new problems of the lubrication of high-speed bearings which require fresh illumination. Unfortunately, much of the experimental work which has been done on a laboratory scale has not been sufficiently extended to approach the conditions of modern turbine practice. This remark applies particularly to the range of surface speeds.

Further, the dimensions of experimental bearings are small compared with bearings used in practice and differ sufficiently in respect of arrangement and clearance to give rise to the possibility not only of a " scale effect " but also of different phenomena occurring in the two cases. Past experimental work too has often provided an inadequate guide for turbine practice because of the tendency to experiment with partial bearings subtending arcs of 180 deg. or less instead of the totally enclosed bearing.

Several turbine manufacturers have done something to fill in the

gap by experiments of their own, covering the range of work in which they are interested. It is not without interest to compare the predictions of some of the outstanding analytical work of various authorities with the results of some such typical tests taken under controlled conditions.

Fig. 1 shows the results of two typical series of tests carried out by the Mechanical Engineering Department of the authors' firm on two types of large turbine bearings in which the maximum test load was 300 lb. per sq. in. of projected area, the maximum rubbing speed was 216 ft. per sec., and the oil used had a viscosity at 120 deg. F. of 0·172 C.G.S. units. Fig. 1 emphasizes the wide variation in loss predicted by different authorities both as between themselves and as compared with the test results. Bearing "A" was a normal type cylindrical bearing without any special features, while bearing "B" was a special bearing in which provision was made to replace the oil by fresh oil after it had been circulated once round the journal. The difference exhibited between the two tests emphasizes the great variation in performance which can be obtained by changing the geography between the journal and the whitemetal which surrounds it.

Experiments made on large-scale high-speed commercial bearings point to the possibility of considerable variation in loss in the bearing with the quantity of oil supplied to it. This variation in loss is far greater than can be accounted for by the change in oil viscosity which accompanies an increase in oil quantity. In these tests it was found that on bearing "B" the loss changed from 60·4 kW. when the bearing was supplied with 13·75 gal. per min. to 109·1 kW. with 39·4 gal. per min. In these two tests the mean oil temperature was different only by 1·4 deg. F.; the change in viscosity at mean temperature being only 3 per cent fails to account for some 96 per cent of the observed increase in power when the oil quantity is trebled. This variation of loss with oil quantity deserves systematic investigation.

There are very definite limits to the permissible reduction in oil quantity for a particular high-speed bearing. As the oil quantity is reduced a point is reached at which vibration begins to be experienced as a result of some instability in the oil film, the temperature of the whitemetal near the point of maximum approach begins to vary rapidly along the length of the bearing and its maximum value rises rapidly compared with the mean outlet oil temperature, while the degree of aeration of the oil leaving the bearing rapidly increases. Further, wiping of the whitemetal may occur. It is obvious therefore that in commercial work the quantity of oil supplied to a bearing must be very considerably greater than the minimum with which the bearing can be run under controlled conditions.

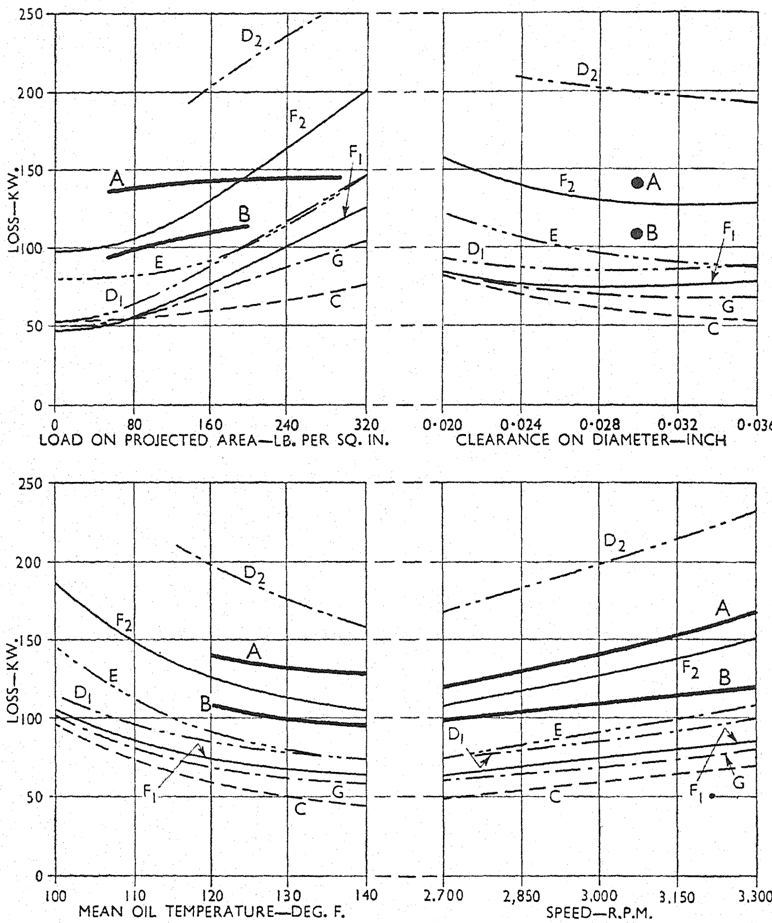

Fig. 1. Comparison of Calculated Losses with Tests

15 × 22-inch experimental bearing, 3,000 r.p.m.

Standard Conditions. Load 150 lb. per sq. in., mean oil temperature 120 deg. F., clearance 0·030 inch in diameter, speed 3,000 r.p.m. For each test or calculated point all conditions except one were kept standard.

Bearings Tested. A, Standard-type bearing, 15 inches in diameter, 22 inches long. B, Scraper-type bearing, 15 inches in diameter, 22 inches long.

Authorities for Calculations:—

C. Sommerfeld, A. *Zeitschrift für Mathematik und Physik*, 1904, vol. 50, p. 97. 180 deg. bearing without side leakage correction.

D1. Hummel, C. Forschungsarbeiten, 1926, No. 287. 180 deg. bearing.

D2. Hummel, C. Forschungsarbeiten, 1926, No. 287. 360 deg. bearing.

Continued on next page

Incidentally, it should be mentioned that in assessing the merits of particular types of bearings and their lubrication, the difference in temperature between the outlet oil and that of the hottest point in the whitemetal is a criterion of much importance. In fact, since it may control safety, it may outweigh in importance questions of securing minimum bearing losses.

Most experienced turbine designers have encountered cases of vibration which arise from some instability in the oil film of the bearing. The vibrations can be as violent as those experienced with a badly balanced machine or with one running near a critical speed. The conditions appear to be readily encountered with lightly loaded bearings with pressures below 40 lb. per sq. in. of projected area and particularly on shafts with lightly loaded bearings running above the first critical speed. While the phenomenon is not at present fully understood, designers have found several expedients by which the trouble can be remedied in particular cases.

There are still many problems to be elucidated by co-ordinated research, but to be successful such research would require to be more extensively planned than is possible to independent experimenters. The experiences referred to indicate that this experimental work must be carried out over and beyond the full range of loadings and speeds that practice now requires or may require in the near future. It will also be necessary to work with complete bearings, with rotors whose stiffness can be varied running in the bearings and with arrangements by which the stiffness of the supporting pedestals can be varied. The characteristics of bearings as suction pumps deserve exploration as do also the effects on stability of the oil film of the ingress of air in the vacuum portions of the film.

The authors are indebted to the following manufacturers of steam turbines who have co-operated in supplying the information on which this summary is based: British Thomson-Houston Company, Ltd., English Electric Company, Ltd., Fraser and Chalmers Engineering Works, Metropolitan-Vickers Electrical Company, Ltd., C. A. Parsons and Company, Ltd., and Richardsons Westgarth and Company, Ltd.

E. Gümbel, L. "Grundzüge der Schmiertechnik", 1931. 180 deg. bearing, with 1917 side-leakage correction, analysis summarized by Falz, E.

F1. Howarth, H. A. S. Trans. A.S.M.E., 1935, MSP–57–2. 180 deg. bearing.

F2. Howarth, H. A. S. Trans. A.S.M.E., 1935, MSP–57–2. 360 deg. bearing.

G. Swift, H. W. Proc. I.Mech.E., 1935, vol. 129, p. 399. 180 deg. centrally loaded bearing with Kingsbury side-leakage correction.

EXPERIMENTAL STUDY OF JOURNAL BEARINGS

By Professor Ch. Hanocq *

PART I. COMPLETE BEARINGS

This paper summarizes a series of theoretical and experimental studies on the full journal bearing which were published between 1929 and 1931.†

Theoretical Deductions. Theory indicates that, in a complete bearing, the centre of the shaft is displaced in a direction perpendicular to the

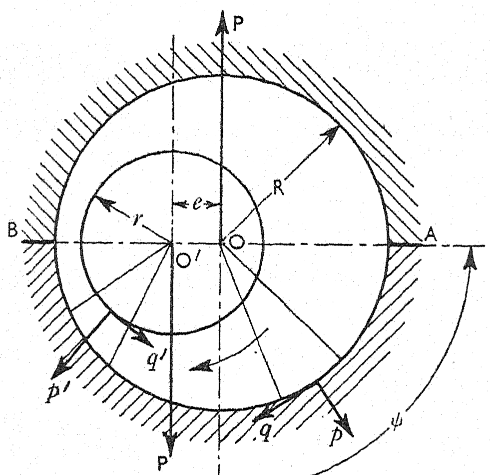

Fig. 1. Displacement of Journal

direction of the load (Fig. 1) as the value of

$$X = \frac{\mu N}{p}\left(\frac{r}{a}\right)^2$$

passes from ∞ to zero. When $X=\infty$, then $e=0$, whereas when $X=0$, then $e=a$. In the above equation μ represents the coefficient of absolute viscosity in kilogrammes, metres, seconds; N is the angular speed in revolutions per second; p is the specific pressure in kilogrammes per square metre; e is the distance between the centres; a the radial clearance $(R-r)$; and c the ratio a/e.

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† See *Revue Universelle des Mines* and Mémoires du Congrès International de Mécanique appliquée, Liége, 1930.

Fig. 2 shows how c varies with X and gives the values of $f_c(r/a)$ and of $f(r/a)$. It can be shown that between f_c and f there is the relation

$$f=f_c+\frac{e}{r} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

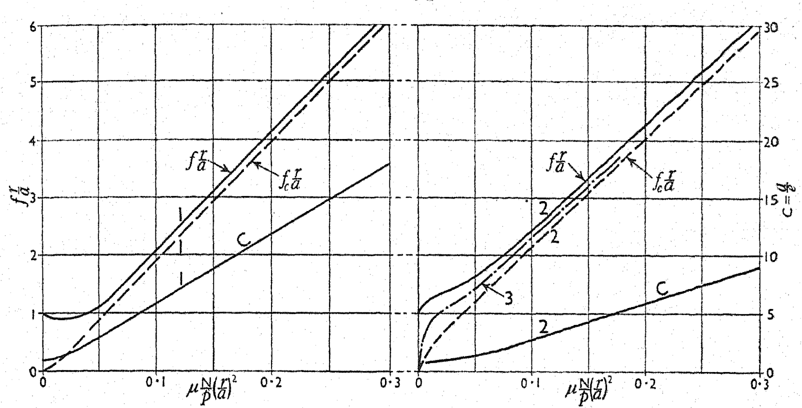
since, if we neglect the forces q in relation to p (Fig. 1), then

$$\mathrm{M}=\mathrm{M}_c+\mathrm{P}\times e \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

The theoretical deduction regarding the value of c and thus of $(f-f_c)$ only holds for very low average specific pressures $\bar{p}$, defined by

$$\bar{p}=\frac{\mathrm{P}}{l\times d} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

This theory is based on the hypothesis that a continuous film



Case 1 Case 2

Fig. 2. Variation of c with X

surrounds the whole shaft. If we try, under these conditions, to deduce the value of the pressure at any point, or, preferably, the value of the ratio $p/\bar{p}$ for different values of X, it is found that:—

(1) The distribution depends essentially on the point at which the lubricant is introduced into the bearing;

(2) When the oil is introduced on the upper surface or on the horizontal diameter negative pressures may exist under the upper half-brass, i.e. pressures less than that of the atmosphere. As these pressures cannot be greater than 1 kg. per sq. cm., there is an average limiting value for the mean pressure $\bar{p}$ beyond which the theoretical conditions are no longer fulfilled.

(3) If it is assumed that atmospheric pressure prevails over the whole of the upper half-brass, the distribution of the pressures $p/\bar{p}$ can be

expressed mathematically. For X=0·013, the values of $p/\bar{p}$ corresponding to this assumption are given in Fig. 3. Here again the pressures are negative and for X=0·013, $p/\bar{p}$ attains a minimum value of 4, i.e. with $\bar{p}$=0·25 kg. per sq. cm., p will be 1 kg. per sq. cm. This equilibrium in turn becomes impossible beyond a certain value of the average pressure $\bar{p}$; atmospheric pressure tends to become established on the arc BC and equilibrium between the load P and the elementary forces $prd\alpha$ is set up only on the arc AC.

The variation of c and of ψ in terms of X depends thus on the way in which atmospheric pressure is established on part of the periphery of the upper half brass; in other words it depends finally on X and on $\bar{p}$.

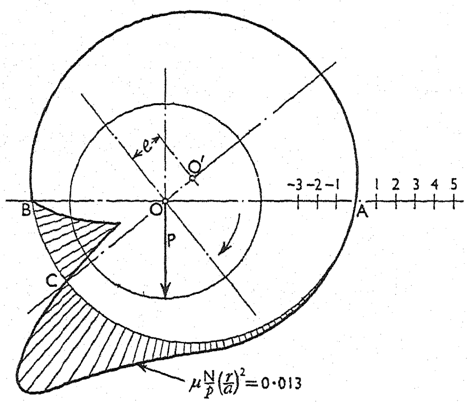


Fig. 3. Values of $p/\bar{p}$

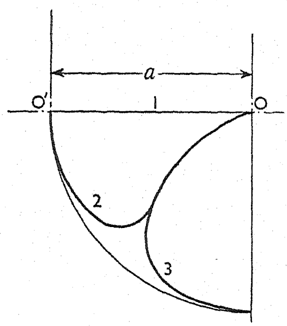


Fig. 4. The Three Limiting Cases

Fig. 4 summarizes the three limiting cases of the displacement of the centre of the shaft in relation to the centre of the bearing:—

Case 1. Complete bearing, continuous film, low pressure:

$$\psi=\frac{\pi}{2}$$

Case 2. Complete bearing, but with a continuous film only on the lower half brass.

Case 3. Complete bearing, with a continuous film on the arc AC only.

As the value of f is always

$$f=f_c+\frac{e}{r}\sin\psi \qquad \ldots\ldots\ldots \quad (4)$$

the difference between f and f_c will, other things being equal, always be less in cases 2 and 3 than in case 1, since $e \sin \psi$ tends finally to become zero in case 3 whereas in case 2 $e \sin \psi$ tends towards the value a as in case 1.

We can thus anticipate that conditions will become unstable when $\bar{p}$ is varied or, even when p and N are constant, X decreases, owing to the decrease in viscosity μ with the temperature. Fig. 2 gives the values of c and of $f(r/a)$ and $f_c(r/a)$ according to case (1), in comparison with the same values deduced according to case (2).

Experimental Study. The method employed was designed to enable the values of M and M_c to be determined simultaneously, so that f, f_c, and $e \sin \psi$ could be determined.

To determine M a decelerating method was employed; a shaft supported by two similar bearings and loaded with two flywheels turns at a speed N; by determining N at different times t, the deceleration curve can be deduced, giving N or ω in terms of t. From this curve we can then deduce $d\omega/dt$ for different values of N and therefore M, since

$$2M + M_1 = -I\frac{d\omega}{dt} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

where M_1 represents the moment of resistance due to atmospheric friction on the flywheels (which can be calculated as a function of their dimensions and speed ω).

The direct method was used for M_c, the bearing being carried on two ball bearings so that it was free to rotate about its axis. A counterweight p applied at a distance λ from this axis, enabled the bearing to be kept in its initial position, whence $M_c = p \times \lambda$.

To determine f and f_c at high loads, ball bearing mountings were employed and the coefficient of friction of the ball bearings was investigated in order to calculate the couple M_r due to them, as this couple affects equation (5). In this way f and f_c were determined for values of $\bar{p}$ ranging from 2·7 to 20 kg. per sq. cm.

The experimental results (Fig. 5) show that the points for values of $\bar{p}$ less than or equal to 5 kg. per sq. cm. fall approximately on curve 2; the points for higher pressures are more regular and lie on curve 3. As the experimental results give a curve for f_c which corresponds to the theoretical curve, we can deduce f in terms of X. The results are given in Fig. 6, where curve 2 refers to pressures equal to or below 5 kg. per sq. cm., and curve 3 to pressures above 5 kg. per sq. cm.

All these results were obtained with a clearance $a/r = 1/170$, using a fixed oil-ring bearing, the oil being admitted at the upper surface.

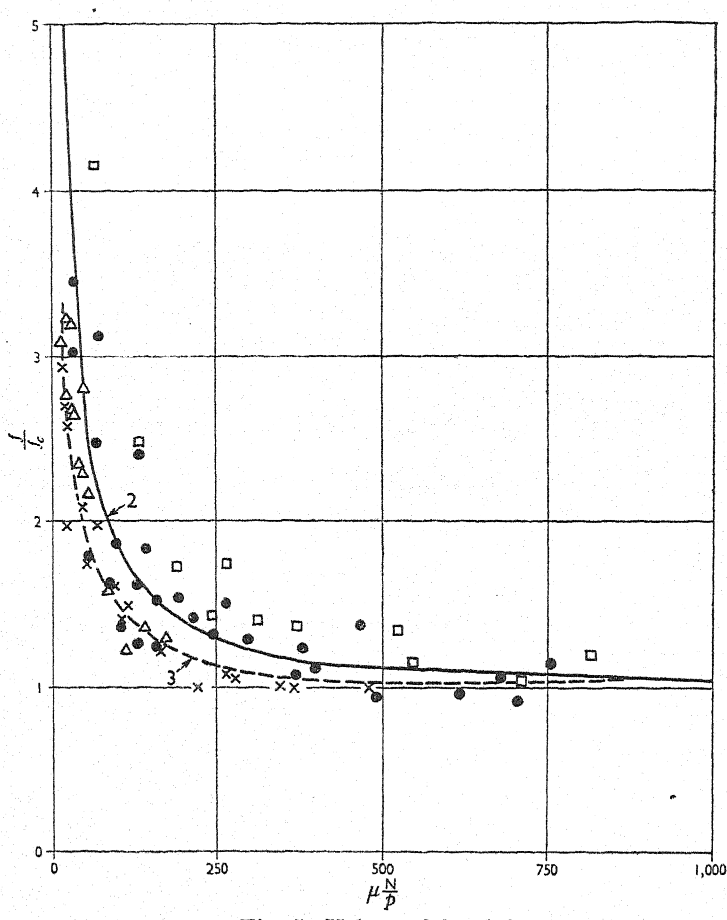


Fig. 5. Values of f and f_c

$2r=40$ mm. $\frac{a}{r}=\frac{1}{170}$

□ $p=$ 2·7 kg. per sq. cm.
● $p=$ 5 ,, ,,
× $p=10$,, ,,
△ $p=20$,, ,,

Conclusions

The author's and certain American investigations show that theory is fulfilled in a remarkable way and that the curves of Fig. 2 suffice to give the value of f. Curve 2 is suitable for low pressures and curve

3 for high pressures, provided a complete journal bearing is used which is lubricated at the upper surface under atmospheric pressure.

Two points should be noted:—

(1) At values of $(\mu N/p) \times 10^8$ less than, say 5, the film ceases to exist, and the equation for f changes completely, as f rises very rapidly with the decrease of X ;

(2) With values of a/r below, say, 1/500, the value of f increases by a constant amount independently of X, varying with a/r in such a way that

$$f_t = f + \Delta f \text{ and } \Delta f = 10^{-13} \times 2{\cdot}66\left(\frac{r}{a}\right)^3 \quad . \quad . \quad . \quad . \quad (6)$$

f being derived from the curves of Fig. 2.

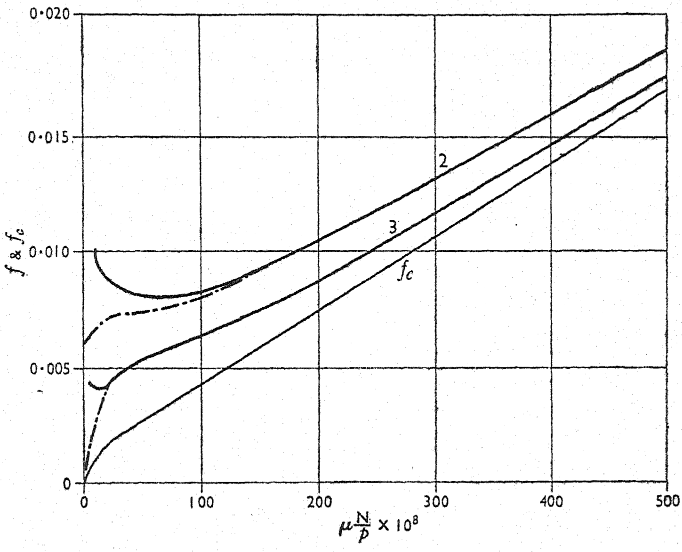


Fig. 6. Values of f in terms of X

Curve 2 Pressures equal to or below 5 kg. per sq. cm.
„ 3 „ above 5 kg. per sq. cm.

The critical value of $(\mu N/p) \times 10^8$ is dealt with below in connexion with partial brasses and here it suffices to say that the limiting value can fall below 5, particularly when the pressures are not too high, say 10 kg. per sq. cm.

All the curves hold good for ratios of l to d above 0·8 ; at lower values, f increases appreciably, as lateral leakage reduces the thickness of the

oil film. So long as hydrodynamic conditions are fulfilled, i.e. for all values of

$$\frac{\mu N}{p} \times 10^8 \geqq 5$$

the coefficient of friction f *depends neither on the chemical nature of the oil nor on the composition of the bearing metal.*

To prove that the nature of the oil has no influence the author used as lubricant a sugar syrup of known viscosity. The values of f_c and therefore of f were given by curves identical with those obtained using oil as lubricant.

Part II. Partial Bearings

The mathematical theory summarized in Part I is not applicable to journal bearings with partial brasses (Fig. 7). For small angles (2β) the bearing can be regarded with fair accuracy as an articulated block, for which the equation for f in terms of the independent variable $\mu N/p$ takes the form $f = A\sqrt{\frac{\mu N}{p}}$.

Boswall has obtained values of A for different angles 2β and has made it possible to measure not only the couple M but also the value of M and therefore

$$f - f_c = e/r \sin\psi = \tan\alpha.$$

With brasses having a relative clearance $a/r = 1/250$, Boswall obtained points falling on different curves according as the bearing angle was between 45 and 90 deg. or less than 30 deg. The approximately parabolic axes in Fig. 9 are reasonable extrapolations of these curves for high pressures.

The author decided to determine the coefficient of friction f for high pressures such as are usual in practice. The first requirement was to determine the equation for f in terms of $\mu N/p$ for brasses with an angle 2β becoming smaller and smaller. To avoid as far as possible the need for measuring $(f - f_c)$ brasses run-in in the cold were employed.

The testing machine employed by the Société Générale Isothermos of Paris, which is based on the principle of the balance applied to the brass (Fig. 8), is designed to work with ordinary journals having a diameter of 140 mm. and an axial width of 300 mm. The load could be increased to 12,000 kg. and the speed to 840 r.p.m. By measuring the reaction R opposed to the force F, applied at M and

resulting from the action of F, the couple $F \times r = R \times L$ can be deduced. Therefore

$$f = \frac{L}{r} \cdot \frac{R}{P}$$

Thanks to the accurate finish of the knife-edges, the coefficient of friction could be measured with an accuracy of roughly 1 in 10,000.

So as to eliminate the error arising from the difficulty of ensuring that the force P acts precisely along the vertical passing through the

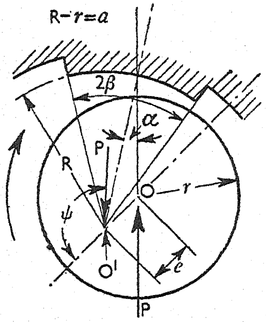


Fig. 7. Journal Bearing with Partial Brass

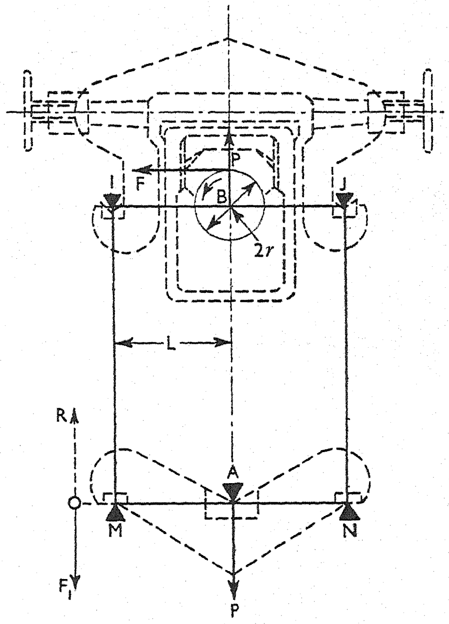


Fig. 8. Diagram of Testing Machine

axis of the shaft, double readings were taken, the working direction being reversed. As the upper rocking lever could be displaced with respect to the brass by hand adjustment of one of the wheels on the right and left of the rocking lever (Fig. 8), the line of action of the load could be moved so as to pass through the centre of the shaft. In this case the readings taken in both directions of running gave practically the same result, since

$$R_1 = F_1 = \frac{F \times r}{L + e^1} \qquad R_2 = F_2 = \frac{F \times r}{L - e^1}$$

Mr. Bastin, director of the Isothermos Laboratory, fitted the rocking lever with a very sensitive water level and a comparator to measure the displacement of the lever with respect to the brass, so that the system could be balanced without adding a counterpoise. Thus if the axis of the applied force is moved to a distance E from the vertical passing through the centre of the shaft so that the rocking lever remains

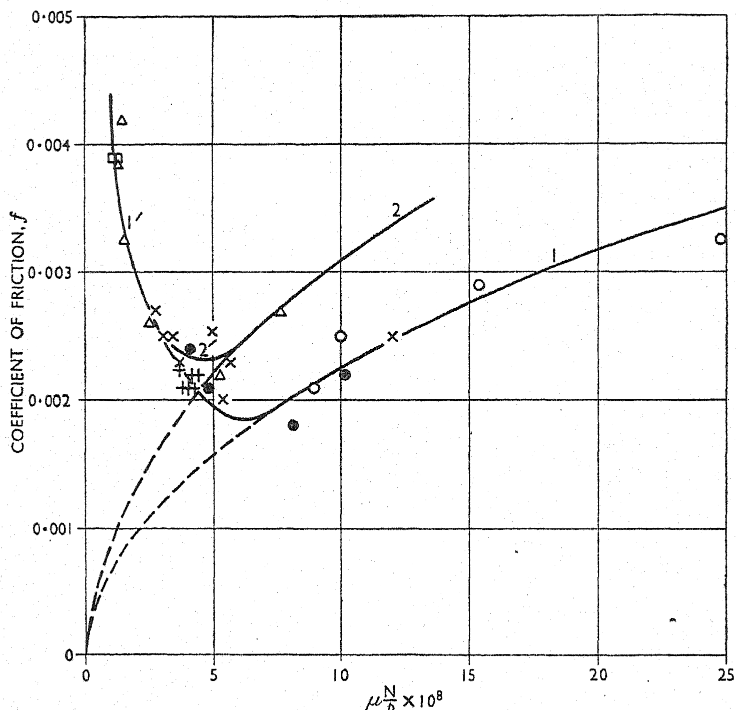


Fig. 9. Curves for Different Bearing Arcs

○	102 deg.	×	41 deg.
●	49 ,,	△	33 ,,
+	45 ,,	□	24·5 ,,

horizontal, the system can be said to be in equilibrium, so that $f=F/P=E/r$.

The results with brasses run-in in the cold are given in Fig. 9. It is noteworthy that curve (1) (for angles above 45 deg.) agrees strictly with Boswall's curve. For angles below 45 deg., the curve (2) traced by the author is the same as that given by Boswall for angles equal to or less than 30 deg.

Therefore the conclusions that A=7·10 for values of 2β between 90 deg. and 45 deg., and A=8·80 for values of 2β equal to or less than 30 deg. can be extended to small values of $\mu N/p$.

There is, however, a critical value of $\mu N/p$ beyond which the expression changes completely, and this is of prime importance in practice. If it is important to approach the minimum of f more closely, it is still more important not to run the risk of attaining a value of $\mu N/p$ below

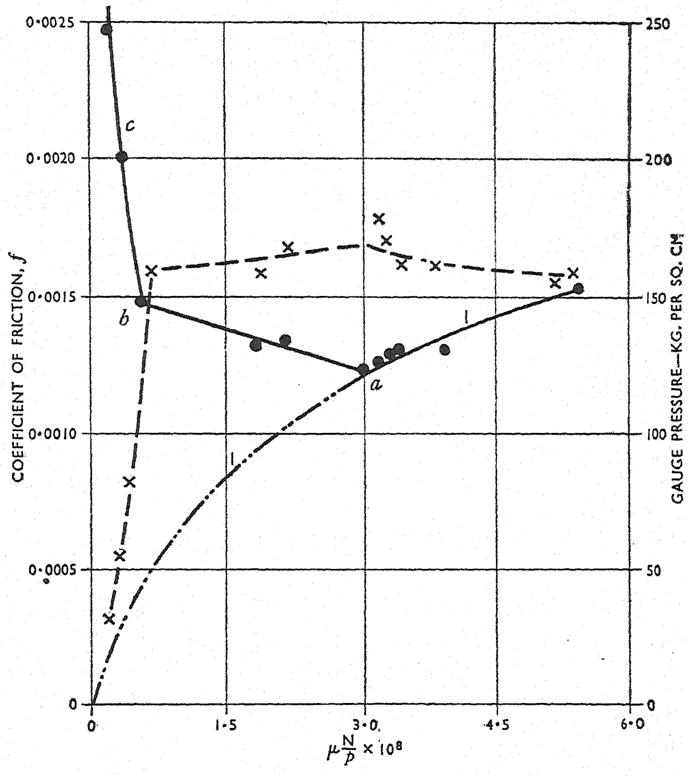


Fig. 10. Critical Value of $\mu N/p$

the critical value as μ becomes less, following on a temperature rise in the bearing. As this question is so important, the Isothermos Laboratory has attempted to ascertain where the danger point lies.

By placing a pressure gauge on the upper surface of the brass, and working with angles 2β=60 deg., such as are common in practice, and clearances of the order 1/50, results were obtained which are reproduced in Fig. 10. The load was kept constant at 7,000 kg. and

the speed was varied. The specific pressure p was calculated from

$$p=\frac{P}{2rl\sin\beta}$$

2β being the effective angle subtended by the brass.

The curve of f retains its parabolic form down to the abscissa 3; with lower values of f it slowly increases and only below 0·7 does the phenomenon change completely, greasy friction replacing hydrodynamic friction. This change in conditions is detected by the pressure gauge, which, up to (*a*) continues to register pressures of 150 to 174 kg. per sq. cm., and after (*b*) shows pressures falling rapidly towards zero.

This result is remarkable: the film holds under a load of 7,000 kg., while the revolutions fall to 4 per minute. At $1\frac{1}{2}$ r.p.m., the coefficient of friction has not yet become twice the minimum coefficient of friction.

Conclusions

The following conclusions have been arrived at:—

(1) The coefficient of friction f can decrease below 0·002 and may become 0·0015 under certain conditions.

(2) Hydrodynamic conditions are established almost instantaneously thanks to the oil held in the clearance space between brass and shaft. The experiments of the Isothermos Laboratory confirm Goodman's work in this respect and show that at speeds above 3 to 4 r.p.m. greasy friction ceases.

(3) To avoid the danger of falling below the critical point, the oil must be selected so that, at the temperature attained by the bearing, the coefficient of absolute viscosity, expressed in kilogrammes, metres, and seconds, will lead to a value of

$$\mu N/p\times 10^8 \geqq 3$$

THE NATURE OF LIMIT FRICTION

By Professor E. Heidebroek*

Usually the coefficient of friction μ decreases with decreasing journal velocity to a minimum, then increases sharply as the velocity continues to decrease. Fig. 1 gives values of $\mu = R/P$ obtained in the author's investigations on phenol plastic bushes, R being the tangential resistance to thrust at the shaft circumference and P the external load.† The region of velocities lying below μ_{min} is that of limit friction or semi-dry friction, and it is supposed that in this region the hydrodynamic laws governing friction no longer hold good. As these laws always hold good under the most varied conditions of the sliding and rolling friction of lubricated surfaces, the contrary supposition is all the less acceptable in that recent researches‡ have shown that lubricating films of a thickness less than 0·001 mm. down to molecular dimensions can exist, the oil molecules being orientated longitudinally. It may be that the increase in the value of μ constitutes a special case of the hydrodynamic theory of friction.

The circumferential friction R_o on the journal can be expressed§ by

$$R_o = r_1 \eta U/mc \int_{\phi_2}^{\phi_1} \tfrac{3}{4}k_1 \sin 2\phi + (\tfrac{3}{2}k_1 - 2)\phi \quad \text{kg. per cm} \quad . \quad (1)$$

This expression is related to the friction on the surface of the bearing. If the corresponding value for the surface of the journal is calculated, then

$$R_o = r_1 \eta U/mc \int_{\phi_2}^{\phi_1} 2{\cdot}161\phi - 0{\cdot}9195 \sin 2\phi = r_1 \eta U/mc(K_\phi) \quad . \quad (2)$$

The term K_o is the integral of a trigonometrical function of ϕ between the given limits ϕ_1 and ϕ_2. Thus R_o depends, except for r_1, η, and U, on $1/mc$. With $m = \sqrt{h_o/c}$ then $1/mc = 1/\sqrt{h_o}\sqrt{c}$, and $R_o = f(1/\sqrt{h_o}\sqrt{c})$, where h_o is the least width of the wedge, and c the bending number ("Schmiegungszahl") $= \frac{1}{2}(r_1 r_2 \pm r_1^2)/r_2 = \frac{1}{2}(r_1 \pm r_1^2/r_2)$. Given η, U, and r_1, R_o remains dependent on h_o and c. The external load does not appear in the equation; it only affects h_o when the journal

* Technische Hochschule, Dresden.

† Heidebroek, E., *Kunst- und Prestoffe*, 1937, part 1.

† Donandt, Maschinen Elemente Tagung, 1936, VDI.-Verlag, p. 33.

§ *Forschung*, 1935, vol. 6, part 4, p. 166.

is floating freely in the bearing. When h_o decreases, R_o increases in accordance with $\tau = \eta \, . dU/dy$. The geometrical position for the different attitudes of the journal is given by a semicircle of the diameter $d = R - r$ (Fig. 2). This relation holds for the ideal journal with a perfectly smooth cylindrical surface.

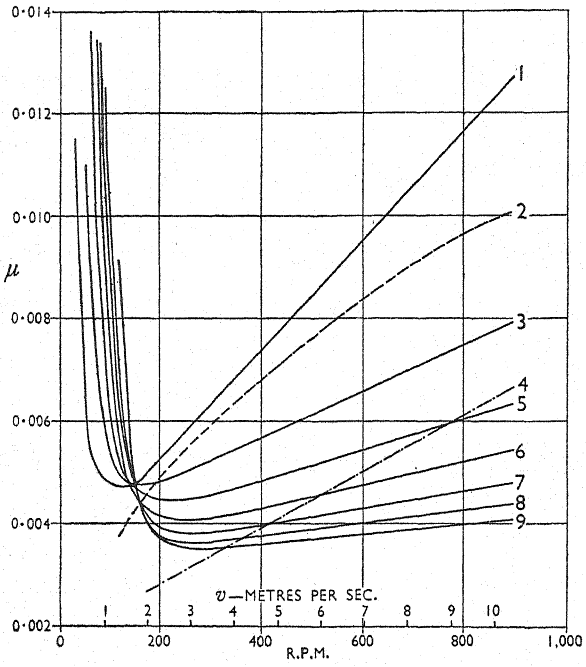


Fig. 1. Tests with Phenol Plastic Bushes

Loads, kg. per sq. cm.

1	1·51	6	6·06
2	4·55 (after Lasche)	7	7·58
3	3·03	8	9·10
4	4·55 (after Rumpf)	9	10·6
5	4·55		

When the surface is uneven, various assumptions can be made. With turned and ground journals or brasses it is probable that the irregularities run in a direction corresponding to the feed of the tool (Fig. 4). Let it be supposed that the journal surface is perfectly smooth, but that the brass shows hills and valleys which run parallel over the circumference. This unevenness has no effect provided that the height of the hills is small compared with the free width of the

wedge. With decreasing values of U, the extent of the pressure increase, $p=6\eta U r_1/m^3c^2 \cdot (0{\cdot}0632-G_\phi)$ also decreases*. Therefore the journal moves downwards on its semicircular path, i.e. h_o becomes smaller, and the load capacity of the oil film diminishes. As the surface of the journal approaches the summits of the hills of the brass, the load capacity again diminishes because the pressure ridge formed on the top of the hills flows into the valleys, so that the pressure now falls near the ends of the brass in accordance with a parabolic law. The surface now has small, separate bearing areas between which run channels through which oil flows back from the area under higher pressure. Here the interconnexion of areas under high pressure with

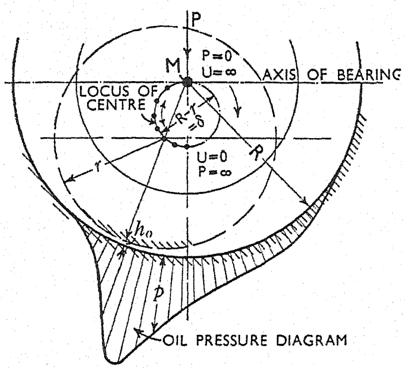


Fig. 2. Attitude of the Journal

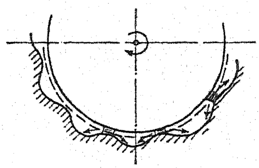


Fig. 3. Effect of Axial Irregularities

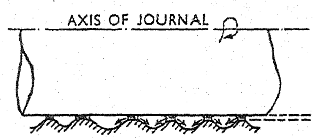


Fig. 4. Effect of Circumferential Irregularities

those of low pressure, a state which is to be avoided in bearings without oil grooves, especially circular grooves, becomes a reality. The oil-pressure ridge is broken down and the load capacity diminishes; in other words, the narrowest wedges on the summits become very small, so that $R_o=(1/\sqrt{h_o}\sqrt{c})U\eta r_1(K_\phi)$ increases considerably. The bending number c remains practically unchanged.

The point where the coefficient of friction μ changes from a falling to an increasing value is to be sought in that displacement of the journal at which the formation of the sliding pressure ridges passes over the upper edges of the irregularities, thus leading to a considerable decrease in the load capacity of the narrowest film. The position of this point, therefore, depends on the surface smoothness which is attainable: the

* *Forschung*, 1935, vol. 6, part 4, p. 163.

more accurate the surface, the smaller the limiting velocity at the transition point.

This explains the fact, which is evident from Fig. 1, that the minimum values of μ all lie close to the same ordinate of velocity, i.e. they lie practically round one point. Then, since, according to equation (2), R_o is independent of the load, the wedge will have, for all loads at a particular value of U, that width at which the bearing film goes

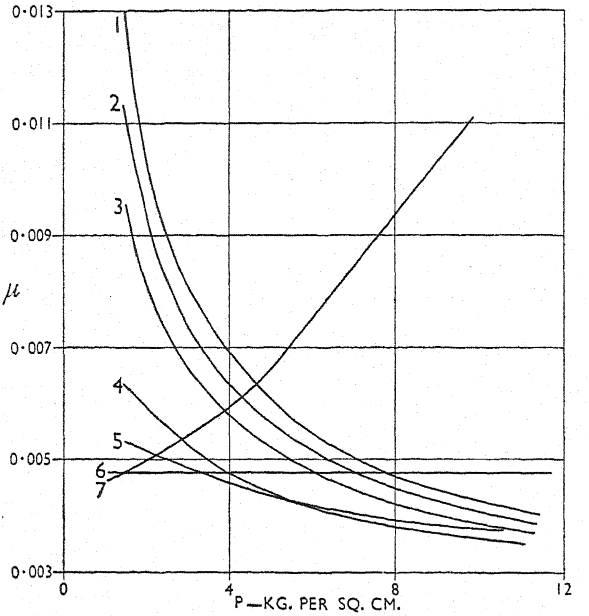


Fig. 5. Friction as a Function of Load

Values of n

1	900 min.$^{-1}$	5	200 min.$^{-1}$
2	750 ,,	6	150 ,,
3	600 ,,	7	100 ,,
4	300 ,,		

over the tops of the irregularities, so that the pressure decreases in the direction of the depressions.

It might be supposed that the irregularities occur in the direction of rotation, i.e. on the circumference of the journal (Fig. 4). This is hardly likely to occur on cylindrical surfaces prepared in the ordinary way, but it is conceivable. In that case, running-in of the journal should result in the formation of numerous very small circumferential

grooves, covering the surface with a network of elevations and depressions. But even apart from this supposition, this situation (Fig. 4) would lead to the continuous pressure ridge in the loaded half of the journal becoming split up into numerous small pressure ridges and so to a reduction in the effective bearing surface. This again implies an increase in the individual partial frictional values. As the passage of oil is impeded, more heat is developed, particularly in the narrowest parts of the wedge, so that the viscosity of the oil and consequently the load capacity are reduced. Then the wedges in the narrowest parts decrease to such an extent that the oil pressure at points of contact so increases as to flatten out the elevations of the bearing metal (this is what occurs during running-in).

The behaviour of the coefficient of friction at small velocities appears particularly clear when the value of μ is expressed as a function of the journal load (Fig. 5). There results a definite limiting velocity at which μ retains the same value for all loads, i.e. it is independent of the load. This limiting velocity corresponds to the point of intersection of the μ curves in Fig. 1. At lower speeds μ increases with the load and, at larger values of U, becomes smaller with increasing load.

Conclusions. The increase in the frictional coefficient μ in the region of limit friction at low sliding velocities is not due to dry friction between the surfaces, but is in accordance with hydrodynamic theory. The shearing force $\tau=\eta(dU/dy)$ depends, according to Reynolds, upon dU/dy, the so-called shear gradient. The thinner the wedge, the larger become dU/dy and τ. Therefore the friction R_o is a function of $1/\sqrt{h_o}$ and depends on the least film thickness and therefore on the position of the journal in the brass. The position of the turning point of the values of μ depends only on the surface irregularity, and thus on the journal attitude at which the width of the free wedge is of so small an order compared with the depressions in the surface that the formation of the oil-pressure wedge passes over the summits of the elevations. Then dry friction (and seizure) do not occur because the frictional values change continually. How far the wedge thickness can decrease is not known, because it depends upon the adsorption between bearing surface and lubricant. The smallest possible values lie under 0·0001 mm. In this respect the new synthetic-resin bearing materials seem to present advantages over the usual bearing metals in respect of clearance and hydraulic limit-friction.

THE PERFORMANCE OF COMPLETE CLEARANCE BEARINGS AS AFFECTED BY CHANGES IN LOAD, SPEED, CLEARANCE, AND LUBRICANT *

By C. Jakeman and A. Fogg, M.Sc.†

The performance of complete clearance bearings is determined by the friction loss and the seizing temperature; in this investigation, the influence on these factors of speed, load, clearance, and lubricant has been examined. Numerous observations, covering a wide range of operating conditions, were made and the complete results will shortly be published. The present paper gives a general survey only of the results obtained and the conclusions drawn.

Range of Investigation and Apparatus. The experiments were made on National Physical Laboratory journal-bearing machines at speeds from 10 to 1,300 r.p.m. and under loads ranging from 180 to 2,500 lb. per sq. in. of projected area. Bronze bushes, 2 inches in internal diameter, were used with diametral clearances ranging from 1 to 16 mils (1 mil=0·001 inch) and the lubricants consisted of two heavy mineral oils, two motor car engine mineral oils, one light compounded machine oil, and castor oil. Particulars of these oils are given in Table 1.

TABLE 1. VISCOSITY OF THE LUBRICANTS UTILIZED AT VARIOUS TEMPERATURES

Oil and description	Viscosity in centipoises at		
	25 deg. C.	100 deg. C.	200 deg. C.
A Light compounded machine oil	55·5	4·2	1·3
B Motor car engine winter-grade mineral oil . . .	325	9·2	1·1
C Heavy mineral oil . .	609	15·4	1·9
D Heavy mineral oil . .	1,178	21·0	1·9
E Castor oil	687	17·4	2·1
F Motor car engine summer-grade mineral oil . . .	462	11·8	1·5

* Work carried out for the Lubrication Research Committee of the Department of Scientific and Industrial Research.

† Engineering Department, National Physical Laboratory.

In the journal bearing machine the bush is loaded by a lever and link system which also provides a means of measuring the friction torque on the bush. The bearing is heated by means of a gas burner from which hot gases pass through the hollow journal into a casing surrounding the bearing. The temperature is measured by a thermocouple placed in a hole drilled into the upper (loaded) side of the bush to within about 0·02 inch of the bearing surface. Oil is pumped to the unloaded side of the bush and, after passing through the bearing, is filtered before returning to the sump. Simultaneous observations of friction and temperature are made from the time of starting the machine at air temperature until seizure occurs. The machine is run with a slack driving belt so that when seizure is approached, the sharp retardation of the journal due to rapidly increasing friction causes the belt to be thrown off the pulley; the temperature at which this occurs is recorded as the seizing temperature. This arrangement acts as a safety device and prevents damage to the bearing surfaces. Considerable care is taken in the preparation of the bearing surfaces as it has been found that small errors in parallelism or circularity and variations in surface finish affect the results to a large extent.

Results. A complete friction-temperature curve was obtained for each condition and the observations have been analyzed to determine whether or not it is possible to deduce any general relation between the variables considered and coefficient of friction and seizing temperature. The analysis indicates that, in many cases, the bushes had not been run for a sufficient period to give consistent observations and later experience has shown that greater consistency would also have been obtained if the filtration of the oil had been more efficient. The results have been divided into three parts, as follows:—

(1) *Effect of Variables on Seizing Temperature.* With all the oils and at all clearances, seizing temperature increased with increase of speed or decrease of load, the rate of increase with speed being much greater at low speeds than at high speeds. Typical values are given in Table 2.

With oils A, B, and C, as the clearance was increased, an optimum value was reached which appeared to depend on the viscosity of the lubricant. The highest seizing temperatures were reached at a clearance of 2 mils with oil A, between 2 and 4 mils with oil B, and at 4 mils with oil C. Comparatively few observations were made with oils D and E because, except at high loads, the seizing temperatures were too high for safe running of the machine. One set of observations which was completed with oil D showed an increase of seizing tem-

TABLE 2. SEIZING TEMPERATURES

Speeds, r.p.m.	40	60	100	150	200	250	300	500	700
Oil A :—									
Load, 180 lb. per sq. in.; clearance, 2 mils. Seizing temperature, deg. C.	59	72	91	113	134	149	162	208	227
Load, 300 lb. per sq. in.; clearance, 2 mils. Seizing temperature, deg. C.	51	65	70	95	111	127	141	180	203
Oil B :—									
Load, 1,000 lb. per sq. in.; clearance, 2 mils. Seizing temperature, deg. C.	42	50	66	81	96	111	125	171	204
Load, 1,000 lb. per sq. in.; clearance, 8 mils. Seizing temperature, deg. C.	22	31	47	60	76	88	102	138	155
Oil C :—									
Load, 300 lb. per sq. in.; clearance, 4 mils. Seizing temperature, deg. C.	78	92	120	150	176	210	225	250	—
Load, 1,000 lb. per sq. in.; clearance, 8 mils. Seizing temperature, deg. C.	60	70	85	100	116	133	149	171	—

perature with speed, although the values were much higher than would have been expected for any of the oils A, B, and C under similar conditions. Very high seizing temperatures were obtained with oil E (castor oil) at low speeds, and chemical decomposition of the oil appeared to be the chief factor determining seizure. Repeated tests on oil F at 1,300 r.p.m. and 1,000 lb. per sq. in. and starting with a new bush, clearance 0·008 inch, gave an increase in seizing temperature from about 200 deg. C. to about 300 deg. C. after five or six seizures. This improvement has been found to be due to modification in the form and surface finish of the bush due to "running-in".

An examination of the values of ZN/P* at seizure indicated that the results had probably been affected to a large extent by running-in of the bushes and that the optimum clearance effect might disappear if the fully run-in condition were reached in each case. Had these conditions been reached it is probable that ZN/P at seizure for the oils A, B, and C would be constant and its value would be about unity (in the units chosen in this investigation). The values of ZN/P at seizure with castor oil were in some cases as low as 0·02 and the large difference between this and the minimum value with the mineral oils indicates that seizing temperature is not determined entirely by the one property, viscosity, of the lubricant, but is also partly dependent on the "oiliness".

(2) *Effect of Variables on Minimum Coefficient of Friction.* Minimum friction occurs between the stage of complete fluid film lubrication (where the friction falls as the viscosity decreases) and seizure. Its actual value is of interest as it gives the minimum friction loss possible under any given set of conditions. The value of the minimum coefficient of friction was found to depend to a certain extent on the initial surface finish given to the bush and journal and on the period of running-in. With bearing surfaces initially polished to a uniform fine finish with the finest grades of emery paper, the minimum coefficient of friction usually decreased slightly as the speed increased, the lowest value with all the oils tested being about 0·0005 to 0·0006. It was unaffected by changes in load or clearance. Repeated tests on oil F at constant speed and load and with the same bush showed a small reduction in minimum coefficient of friction during the first three or four tests, after which it remained constant at a value of $0{\cdot}0005_5$. Very slight local roughening of the bearing surfaces due to accidental passage through the oil film of a small piece of grit—not large enough to be removed by a fine gauze filter—caused an increase in the minimum friction; this was presumably due to high point contact being reached at a lower temperature and thus at a higher viscosity.

It appears from these observations that, with completely run-in bearings, the minimum coefficient of friction would be independent of speed, load, clearance, and lubricant.

(3) *Relation between Coefficient of Friction and* ZN/P. In these experiments the form of the bushes has undergone small continuous changes as the ranges of speed and load have been covered, probably

* Z = viscosity of lubricant in centipoises
N = speed of journal in revolutions per minute.
P = load in pounds per square inch of projected area.

due to wear at seizure, compression of the bush material, and heat distortion. All variables therefore have not been under control and so it has not been possible to obtain a general relation between the coefficient of friction and the parameter ZN/P; information has, however, been obtained on the process of running-in, a process which must take place in practice and which has been accelerated by seizure in this investigation.

In spite of the bush variations, when the coefficient of friction was plotted against ZN/P for each clearance and each oil, the results were sufficiently well grouped to allow reasonable mean curves to be drawn. It should be noted that the values of ZN/P were low and represented

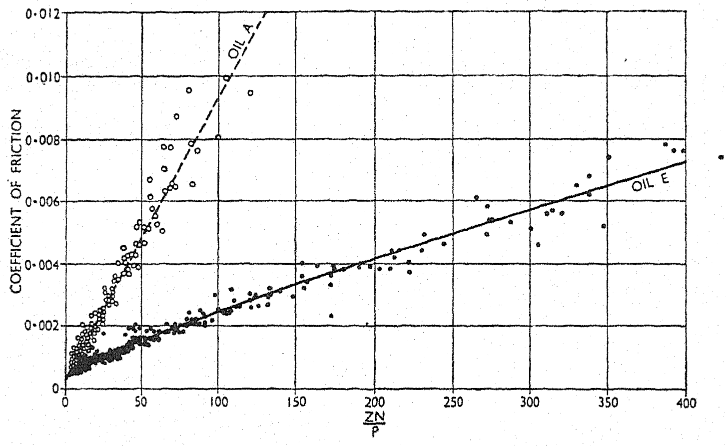


Fig. 1. Coefficient of Friction–ZN/P Curves for Oil A (clearance 0·001 inch) and Oil E (clearance 0·004 inch)

mainly conditions of thin fluid film lubrication where slight variations in the surface condition of the bush and journal would have a considerable influence. Two typical sets of observations are reproduced in Fig. 1, coefficient of friction being plotted against ZN/P and mean curves drawn. Similar curves have been drawn for each clearance and each lubricant; the equations to the mean curves are given in Table 3.

At 1 and 2 mils clearance there is a linear relation between μ and ZN/P for all the oils, whilst for clearances above 4 mils, the index of the factor ZN/P is less than unity. Oils A, B, and E at 1 mil clearance have similar equations, as also have the groups A, C, D, and E at 2 mils clearance and A and C at 4 mils clearance; and the constant b in the product b(ZN/P) is nearly inversely proportional to the clear-

TABLE 3. EQUATIONS TO THE MEAN CURVES

Clearance, mils	Oil	Equation to mean curve $\mu=a+b(\mathrm{ZNP})^n$
1	A	$\mu=0{\cdot}0003\ +9{\cdot}1\times10^{-5}\ \mathrm{ZN/P}$
	B	$\mu=0{\cdot}0003\ +8{\cdot}9\times10^{-5}\ \mathrm{ZN/P}$
	E	$\mu=0{\cdot}0003\ +8{\cdot}2\times10^{-5}\ \mathrm{ZN/P}$
2	A	$\mu=0{\cdot}0004\ +4{\cdot}7\times10^{-5}\ \mathrm{ZN/P}$
	B	$\mu=0{\cdot}0004\ +6{\cdot}0\times10^{-5}\ \mathrm{ZN/P}$
	C	$\mu=0{\cdot}0003_5+3{\cdot}3\times10^{-5}\ \mathrm{ZN/P}$
	D	$\mu=0{\cdot}0005\ +3{\cdot}5\times10^{-5}\ \mathrm{ZN/P}$
	E	$\mu=0{\cdot}0005\ +3{\cdot}4\times10^{-5}\ \mathrm{ZN/P}$
4	A	$\mu=0{\cdot}0005\ +1{\cdot}9\times10^{-5}\ \mathrm{ZN/P}$
	B	$\mu=0{\cdot}0005\ +4{\cdot}7\times10^{-5}\ (\mathrm{ZN/P})^{0{\cdot}86}$
	C	$\mu=0{\cdot}0005\ +1{\cdot}6\times10^{-5}\ \mathrm{ZN/P}$
	D	$\mu=0{\cdot}0004_5+7{\cdot}1\times10^{-5}\ (\mathrm{ZN/P})^{0{\cdot}72}$
	E	$\mu=0{\cdot}0004_5+3{\cdot}7\times10^{-5}(\mathrm{ZN/P})^{0{\cdot}87}$
7	E	$\mu=0{\cdot}0005\ +7{\cdot}0\times10^{-5}(\mathrm{ZN/P})^{0{\cdot}66}$
8	B	$\mu=0{\cdot}0006+3{\cdot}0\times10^{-5}\ (\mathrm{ZN/P})^{0{\cdot}85}$
	C	$\mu=0{\cdot}0005+6{\cdot}5\times10^{-5}\ (\mathrm{ZN/P})^{0{\cdot}68}$
	A	$^*\mu=0{\cdot}0005+4{\cdot}8\times10^{-5}\ (\mathrm{ZN/P})^{0{\cdot}84}$
	E	$^*\mu=0{\cdot}0005+4{\cdot}3\times10^{-5}\ (\mathrm{ZN/P})^{0{\cdot}80}$
	F	$^*\mu=0{\cdot}0005+4{\cdot}8\times10^{-5}\ (\mathrm{ZN/P})^{0{\cdot}80}$
16	C	$\mu=0{\cdot}0006_5+2{\cdot}1\times10^{-5}\ (\mathrm{ZN/P})^{0{\cdot}79}$

* The observations giving these mean curves were obtained on bushes which had previously been run in, i.e. until successive tests gave the same friction temperature curve.

ance, as required by theory. The tests on run-in bushes gave plotted points much nearer to the mean curves than the other tests, indicating that the scatter of points was due chiefly to a gradual running-in effect during the tests. Despite the variations introduced by changes in the form of the bushes, the results show that the friction, even at very low values of ZN/P, is apparently independent of the nature of the lubricants.

General Conclusion. Although the results have been upset by the changes in the form of the bushes, a general indication of the extent to which seizing temperature and friction are dependent on load, speed, clearance, and lubricant may be given. As regards seizing temperature, increase of speed or decrease of load result in increase of seizing temperature, and an increase in viscosity of the lubricant generally has

the same effect. Since the value of ZN/P at seizure with castor oil is lower than with the mineral oils tested, it appears that seizing temperature depends on the "oiliness" of the lubricant in addition to the viscosity. The effect of clearance is masked by the changes in the form of the bushes.

The minimum coefficient of friction would probably be independent of the variables considered with uniformly finished and completely run-in bearings. The actual value of μ_{min} will depend on the quality of the surface finish.

The relation between coefficient of friction and ZN/P can be expressed as $\mu = a + b(ZN/P)^n$, where n appears to be unity for clearances up to 4 mils on 2 inches diameter and lies between 0·65 and 0·85 for greater clearances. For a given value of ZN/P, no appreciable difference between the value of the coefficient of friction with different oils at any one clearance was revealed in this investigation.

END LEAKAGE AND OIL CIRCULATION OF RING-LUBRICATED BEARINGS

By G. A. Juhlin * and R. Poole †

In ring-lubricated bearings, the rings have been designed to supply sufficient oil to maintain a satisfactory oil film and no serious attempt has been made to design from the point of view of heat transfer.

Many tests have been carried out to find the quantity of oil delivered by oil rings and it has been considered part of the design procedure to calculate the quantity of oil leaking from the ends of the bearing in order to ensure that the rings supply sufficient oil to compensate for

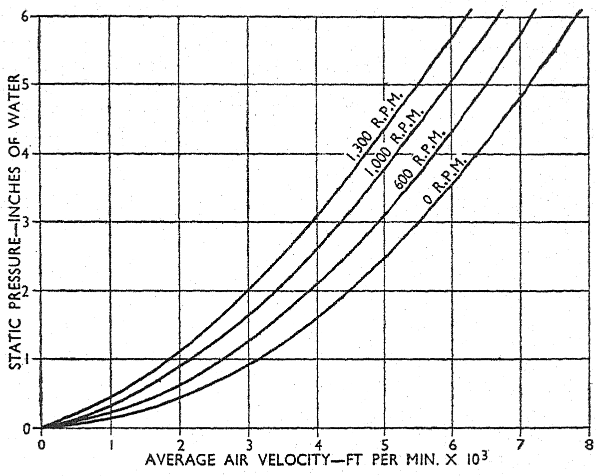


Fig. 1 Total Static Air Pressure Required to Force Air through a ½-inch Single Air Gap with Rotor (Diameter, 25 inches) at Various Speeds

the leakage. Karelitz (1929) has developed an approximate theory for calculating the end leakage with oil-ring bearings. This theory was based on the assumption that "As far as axial flow is concerned, the surfaces of the journal and shell are at rest". In considering the flow of fluid between two co-axial rotating cylinders, Poole (1932) has

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shown that the fluid is rotating with relation to both the outer and inner surfaces. The resistance to flow increases with the relative rotation of the cylinders. This is to be expected, since a particle of fluid moving axially would trace a helix on the rotating surface and a straight line on the stationary surface. Actually, the general mass of fluid is rotating at one-half the cylinder velocity. It has therefore a positive rotation with relation to the stationary cylinder and a negative rotation with relation to the rotating cylinder.

Luke has published results of measurements of the pressure required to force air through the annulus between a 25-inch cylinder rotating in a cylinder of 26-inch bore, the tests being carried out over a wide range of air speed. The results, reproduced in Fig. 1, show a marked increase of resistance with rotation. The axial flow of oil from a journal bearing is much more complicated since the radial gap is not uniform and the pressure varies considerably around the journal. The path of the oil will also be modified by the shape and position of oil grooves.

In spite of these differences, however, it appears reasonable to expect that the journal rotation would increase the resistance of the bearing to axial oil flow. In order to find the effect of speed upon end leakage as calculated by Karelitz's method, the latter's expression has been evaluated for various values of ZN/P. He states that:—

$$Q_{gal}=0{\cdot}00022\ K\frac{Pd}{ZL}(1{,}000S)^3$$

where

- Q_{gal} End leakage in gallons per minute.
- P Pressure in pounds per square inch.
- Z Oil viscosity in centipoises.
- d Journal diameter in inches.
- L Journal length in inches.
- S Bearing clearance, inches on diameter.
- K A constant depending upon α, the angle between the entering edge of the oil film and the direction of load, and the operating value of ZN/P, where N = journal speed in revolutions per minute.

The value of K has been calculated for a clearance-diameter ratio of 0·001 and a clearance-diameter ratio of 0·0025 (Fig. 2).

It is of interest to note that with the small clearance K increases with ZN/P. Hence for a given pressure and viscosity the calculated end leakage increases with journal speed. For the large clearance, however, the increase in the value of K with ZN/P is almost negligible.

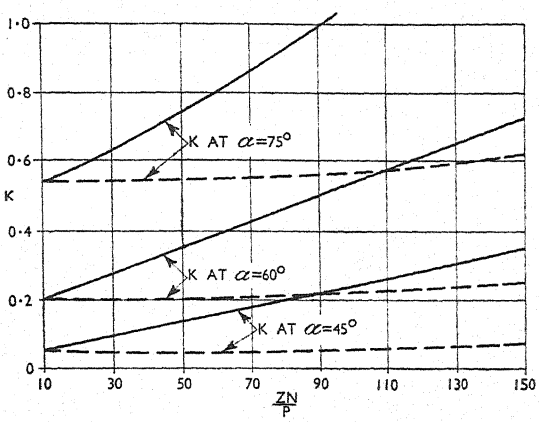
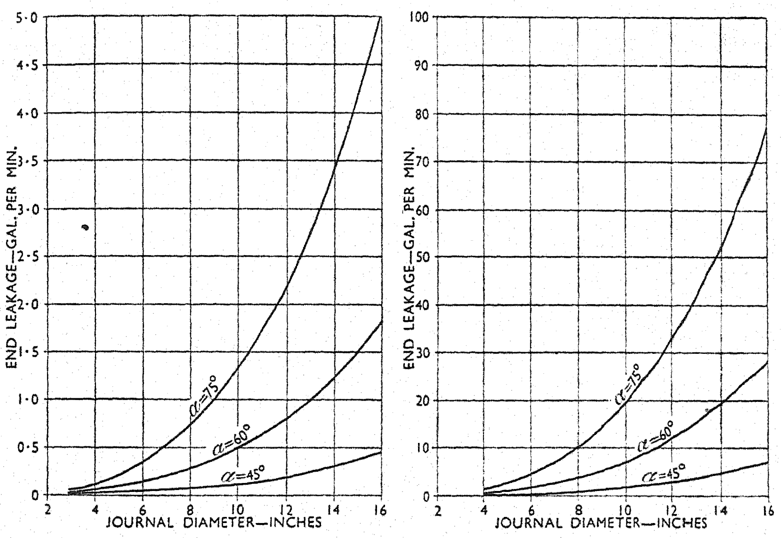


Fig. 2. End Leakage Factor for Journal Bearings, Neglecting the Effect of Shaft Rotation

——— Clearance-diameter ratio = 0·001.
– – – Clearance-diameter ratio = 0·0025.



a *b*

Fig. 3. End Leakage of Journal Bearings

a Length-diameter ratio = 2·0.
Clearance-diameter ratio = 0·001.

b Length-diameter ratio = 2·0.
Clearance-diameter ratio = 0·0025.

Assuming Z=10 centipoises and P=200 lb. per sq. in., curves have been plotted for the end leakage in gal. per min. for the two clearance ratios (Fig. 3 *a* and *b*). For a 75 deg. bearing 16 inches in diameter the estimated end leakage (Fig. 3 *b*) according to Karelitz's method would exceed 75 gal. per min. whilst for a 45 deg. bearing of the same size the end leakage would be 7 gal. per min.

In view of the assumptions in this theory, and in order to check the present authors' opinion that the leakage would decrease with speed, tests were carried out on a small 5-inch by 10-inch bearing having a clearance of 0·001 inch per inch diameter. This bearing was of standard type and no special attention had been paid to it during manufacture. It was used at the commutator end of a two-bearing direct current motor and the load was approximately 50 lb. per sq. in.

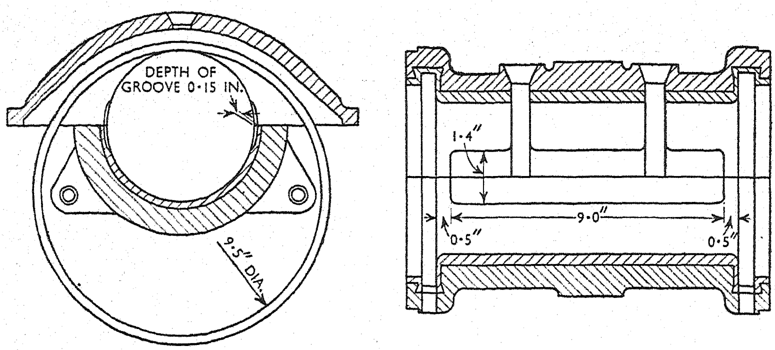


Fig. 4. Details of the Bearing and Oil Rings

At each end of the bearing was a groove to collect the oil running from the ends, and pipes were connected in order that the end leakage could be measured.

In the first series of tests oil rings were fitted; details of the bearing and the oil rings are given in Fig. 4:

Test Results. Fig. 5 shows the relationship between journal speed and oil-ring speed. The oil-ring speed increases with the speed of revolution of the journal and from the numerous tests carried out on oil rings the quantity of oil delivered by the rings would be expected to increase with the speed of the ring.

The variation of measured end leakage with journal revolutions is shown in Fig. 6. It will be noticed that the end leakage increases almost linearly up to 200 r.p.m.; there is then a dip in the curve which afterwards reaches a maximum of 0·042 gal. per min. at a speed of

550 r.p.m. The quantity falls off rapidly as the speed is increased, until at 1,200 r.p.m. the end leakage is approximately 0·01 gal. per min.

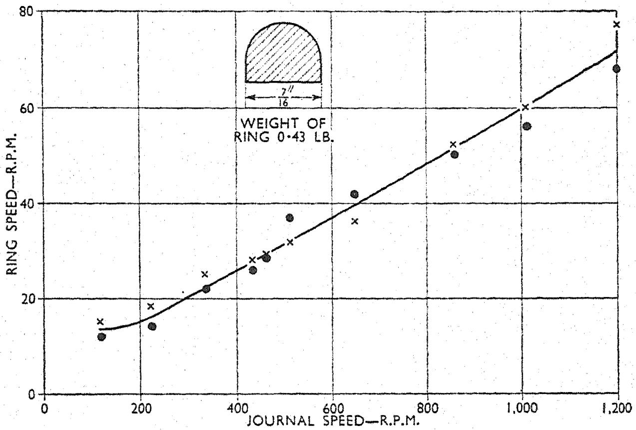


Fig. 5. Relation between Journal Speed and Oil Ring Speed

The Effect of Change in Quantity of Oil Supplied. Since the quantity of oil supplied by the oil rings was unknown, and might vary with

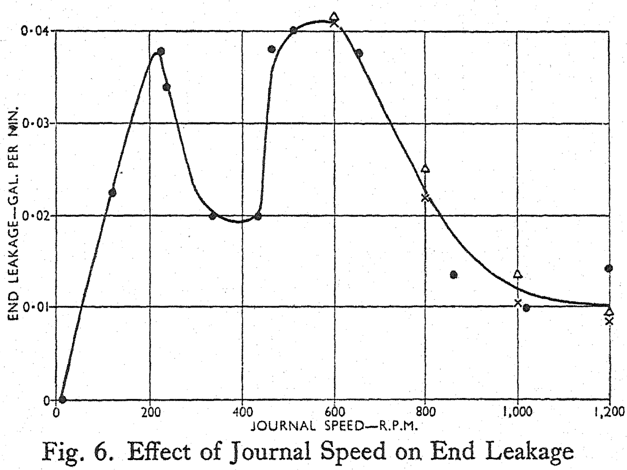


Fig. 6. Effect of Journal Speed on End Leakage

Bearing, 5 × 10 inches.

the speed of revolution of the journal, it was decided to remove the rings and carry out tests with a separate oil supply. Oil, supplied by means of two pipes from a large tank, was allowed to drip into the

bearing at the point where the rings were normally in contact with the journal. It was found that although the quantity of oil supplied was increased to the order of ten times the end leakage, there was no measureable increase in the leakage. The surplus oil overflowed through the oil ring openings in the top half of the bearing.

It is important to note that the relationship between end leakage and speed was for all practical purposes independent of the quantity of oil supplied, provided that the supply exceeded the end leakage. The change in end leakage which occurred at 300 to 400 r.p.m. is therefore associated with the oil film conditions and can in no way be attributed to the variation of the quantity of oil supplied by the oil rings.

The Effect of Change in Oil Temperature. The oil temperature was varied from 47 deg. C. with a corresponding viscosity of 30 centipoises, to 35 deg. C. with a viscosity of 60 centipoises. There was found to be no appreciable change in the end leakage although the oil viscosity had increased by 100 per cent.

Conclusion. The quantity of oil leakage from the ends of a bearing is extremely small and, from the point of view of lubrication, there should be no difficulty in designing oil rings that more than compensate for this end leakage.

Any advantages to be expected from increasing the quantity of oil delivered by oil rings would appear to be attributable to improved cooling of the journal by the excess oil supplied. By paying more attention to the distribution of the surplus oil more efficient cooling may be obtained.

There is undoubtedly a wide field for further investigation of end leakage and the separation of the two functions of oil rings, namely, lubrication and cooling.

It appears from the experiments that existing theories may considerably over-estimate the end leakage from journal bearings, and that extensive experimental work is necessary before any satisfactory method of calculation of end leakage can be evolved.

References

KARELITZ, G. B. 1930 Trans. A.S.M.E., vol. 52, part 2, AM–52–5, p. 57, "Performance of Oil Ring Bearings".

POOLE, R. 1932–3 Tech. Section, Assoc. Eng. and Shipbuilding Draughtsmen, vol. 13, p. 34, "The Design of Fans for Cooling Electrical Machines".

OIL SUPPLY IN SELF-CONTAINED BEARINGS

By Professor George B. Karelitz *

In a bearing with forced-feed lubrication, the amount of oil supplied to the bearing is usually many times the quantity needed actually to maintain a load-carrying oil film. This is not so, however, in "self-contained" bearings, i.e. those which operate without the assistance of external pumps feeding the lubricant into the bearing clearance. This paper only discusses means of oil supply to the shell of such bearings.

Bearings with Perfect Lubrication. Bearings designed for long life are built to operate on a "thick" oil film. The film is wedge shaped, and is from 0·0005 to 0·003 inch thick at the point of closest approach between journal and bearing shell. A hydrostatic pressure is created in the film, the viscous oil being dragged into the converging wedge between the journal and shell. The pressure is sufficient to carry the load on the journal. The film thus separates the rubbing surfaces and prevents their wear most effectively.

As the ends of the bearing open usually into the bearing housing or into grooves, the hydrostatic pressure at the ends is practically zero (gauge). A pressure gradient exists therefore in the axial direction, causing an axial flow of oil through the end clearance, the so-called end leakage. This leakage must be replaced continuously if the load-carrying film is to be maintained. The amounts of oil required for replacement are not large; however, to ensure a margin of safety, the oil supply should be, say, twice the oil loss through end leakage. The axial velocity of the oil is rather small, while the oil is carried along by the rotating journal. The path of a particle of oil entering into the clearance space is therefore helical in shape. Eventually, the particle is carried to the bearing end and thrown out of the clearance. A portion of the end leakage is recirculated, being drawn back into the clearance by the vacuum existing in the divergent wedge of the oil film, beyond the point of nearest approach between journal and bearing shell.

The quantity of oil lost by leakage through the bearing ends was computed for 120 deg. bearings by Needs (1934). Measurements of this leakage on a 120 deg. bearing made (1935) by the author (Fig. 1) compared well with Needs's data. For a wide range of loading conditions (eccentricity from 0·4 to 0·8) and ratios of length to diameter ($\frac{3}{4}$ to $1\frac{1}{2}$), the end leakage may be estimated by the simple formula $Q=0{\cdot}27Ua\eta$, where U is the linear velocity of the journal surface in

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inches per second; a and η are respectively the radius of the journal and the radial clearance in inches; Q is the side leakage in cubic inches per sec.

The common device for lifting the lubricant from the reservoir to the journal is the oil ring. The mechanism of operation of an oil ring is as follows. At very low journal speeds the ring has the same linear speed as the journal, i.e. there is no slip between the two. The oil is lifted by the ring and spread over the journal. The amounts of oil thus delivered at low journal speed are usually far in excess of those needed to make up for the end leakage. At a journal speed of about 200 r.p.m., the ring slips, and instead of a metal-to-metal contact between ring and journal, a viscous oil film builds up between the two. Up to, say, 400 r.p.m. of the journal, the motion of the ring is

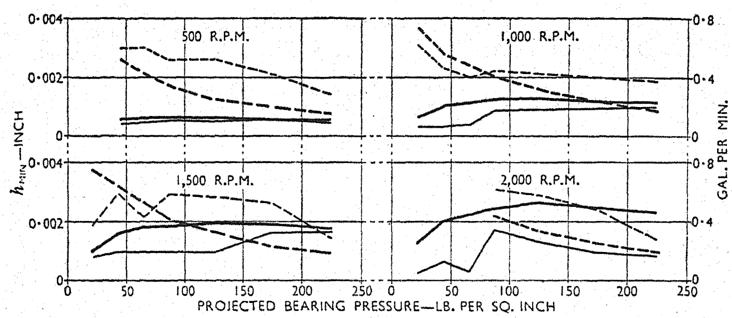

Fig. 1. End Leakage from Journal Bearings

Journal, 6 inches in diameter, 9 inches long.

Calculated oil leakage.
Measured ,, ,,
Calculated oil film thickness.
Measured ,, ,, ,,

unstable; the oil film builds up and breaks down intermittently. At higher speeds of the journal this film is rather stable. The oil ring rides on the film, and is driven only by the viscous friction in the oil film. The resistance of the oil ring to motion consists of the hydraulic resistance of the oil in the reservoir, and of the weight of the oil lifted on the "up" side. Consequently, the speed of the oil ring increases but slightly when the journal is rotated even at a high rate, say 1,800 r.p.m.

An oil film is lifted on the "up" side of the oil ring. The film built up between the ring and journal is rather stable. On the upper part of the ring, the oil is carried as a crown on the ring's outer surface, forced there by the centrifugal force. The oil carried on the inner surface forms the film carrying the oil ring. This oil is squeezed out

from underneath the ring, and coming in contact with the surface of the journal, is thrown off horizontally forward. Two sprays of oil are thus formed on either side of the ring. The cover of the bearing housing, or the bearing shell should be designed so that these sprays are trapped and diverted into the bearing clearance. Otherwise, they will play on to the housing, and the oil will fall uselessly back into the reservoir. It should be noted that most of the oil delivered by an oil ring goes into these two sprays (Karelitz 1930).

Since the amounts of oil carried by the oil ring depend on its speed of rotation, many attempts have been made to increase the driving friction between journal and oil ring. In an effective design due to Baudry and Tichvinsky (1937), a number of circumferential grooves

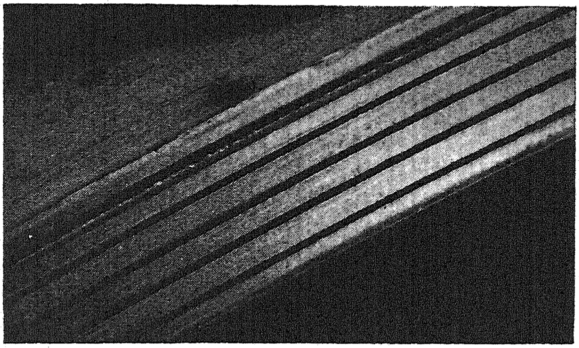

Fig. 2. Oil Ring with Grooved Inner Surface

(Fig. 2) are introduced on the inner surface of the ring, thus thinning out the ring-carrying film. An increase of 50 per cent in oil delivery to the bearing is achieved by this improvement in design (Fig. 3).

Numerous devices have been employed to lift the lubricant from the reservoir, but these are not universal in application. Chains riding the journal, disks mounted on the journal and dipping into the reservoir have been employed. The "Jones" bearing operates successfully under comparatively light loads and at low speeds. In this bearing the lower half-shell is perforated and the lubricant fills the reservoir practically up to the lower line of the journal. The oil is sucked into the bearing by the vacuum in the divergent part of the clearance and lubricates the journal satisfactorily.

Starved Bearings. In spite of the advantages of perfect lubrication with regard to absence of wear of the rubbing surfaces, designers are frequently compelled to avoid this method of lubrication. For instance,

in a grinder with a sleeve bearing, the clearance must necessarily be small. If the clearance were filled with oil, the losses in the clearance, due to the high rate of shear and high speed of rotation, would be excessive. In fact, the motors usually supplied with grinders would hardly be powerful enough to rotate the journals at the rated speed. In this case, the mechanism of the drip-feed lubrication adopted is quite different from that in a perfectly lubricated bearing. The load on the bearing is very small, and the bearing is of considerable length. The journal floats on several puddles of oil, most of the clearance being filled with air. (The clearance being very small it is quite feasible that the air film takes an active part in lubrication.) Under the circum-

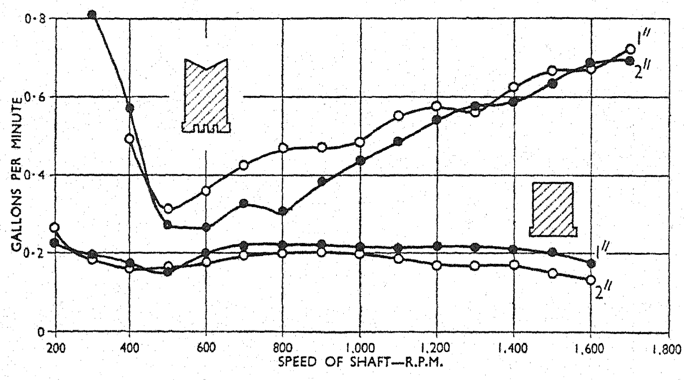

Fig. 3. Comparison of Oil Supply of Standard and Special Oil Rings at 1 inch and 2 inches Submersion

Oil temperature, 60 deg. C.
Diameter of journal, 9 inches. Diameter of oil ring, 15 inches.

stances, there is little leakage from the clearance, the oil circulating inside the clearance from zones of pressure to zones of vacuum. Only stray particles of the lubricant are carried to the bearing ends and thrown off the shaft by centrifugal force. The make-up is therefore small, and a drop feed is ample to maintain lubrication.

The various designs of drop-feed oilers are well known and need not be discussed. The physical phenomena in starved bearings are, however, worthy of study. The number of bearings thus lubricated is great, but the type has been neglected by research workers.

Waste-Packed Bearings. Another deliberate deviation from rational, wear-eliminating, perfect film lubrication is the railway and railway electric motor bearing. Its design was dictated by limitation of space, and the designers resigned themselves to a comparatively rapid wear

of the shells. The purpose of lubrication in railway bearings is to reduce friction sufficiently to keep the temperature of the bearing within reason, and to eliminate seizure and mechanical destruction of the brass or shell. The bearings work distinctly in the semi-fluid range of lubrication. The load is carried partly by the hydrostatic pressure created in the oil film, and partly by direct metal-to-metal contact, where boundary conditions of lubrication take place. Gradual wear and minute thermal distortions continuously change the region of hydrostatic pressure in the bearing, thus varying the circulation of the lubricant in the clearance. However, the general mechanism of operation remains unchanged.

The mechanism of operation of railway bearings has been the subject of numerous investigations in many countries, the economic importance of this bearing being self-evident. For instance, Pearce (1936) has published recently his work in this field.

When the bearing comes to rest, part of the oil circulating in the clearance flows out of the bearing at the ends, the oil being hot and therefore of low viscosity. A certain amount of the lubricant is retained, however, by capillary action, and lubricates the bearing during the first period after the journal has been restarted. After the first few revolutions the journal wipes oil off the waste and the clearance fills up with lubricant. The flow of oil in the clearance depends on its configuration and, like the amount of end leakage, may vary considerably from day to day. At times, the bearing may run for hours with a loss of lubricant amounting to several drops per hour, but at other times the oil loss would be several drops per minute. At any rate, the oil loss cannot be greater than the amount of oil supplied by the waste to the journal, and one of the features of railway bearings is a long period between oilings. It is desirable that the lubricant should be fed sparingly to the journal.

The mechanism of oil supply by the waste has not been given the same attention. The author has investigated this question (1926). The amounts of oil which a wick can syphon depend greatly on the oil lift and on the viscosity of the oil, or its temperature. The surface tension, the capillary action between the fibres of the waste and the specific gravity of the oil change but slightly with temperature. The viscosity of the oil, however, decreases many times with temperature rise and the flow of oil through the wick is substantially inversely proportional to the viscosity of the lubricant. It is therefore natural that some time must elapse between the starting of a cold waste packed bearing and the beginning of lubricant supply. The temperature of the bearing must first come to a point where the viscosity is sufficiently low to permit the oil to flow through the waste.

On the other hand, the performance of the waste-packed bearing does not depend practically on the amount of oil flowing through it, provided that the amount of oil supplied to the journal suffices to keep the clearance between journal and brass reasonably full. It is therefore desirable to use lubricant of high viscosity, and control the oil-lift to minimize the amount of oil fed to the journal where space limitations do not allow the employment of proper oil seals to prevent the loss of oil passed through the bearing.

Conclusion. The recent and sudden general increase in loads, sizes, and speeds of journals in practically every type of machinery has left designers without reliable precedent upon which to base their designs. This necessitates a better qualitative and quantitative knowledge of the existing means of oil supply to bearings, as a basis of further improvement and development of economical bearing design.

References

BAUDRY, R., and TICHVINSKY, L. M. 1937 *Mechanical Engineering*, vol. 59, p. 89, "Performance of Oil Rings".

KARELITZ, G. B. 1926 Trans. A.S.M.E., vol. 48, p. 1165, "The Lubrication of Waste-packed Bearings"; 1927–28 Trans. A.S.M.E., vols. 49–50, APM–50–1, "An Investigation of the Performance of Waste-packed Armature Bearings".

KARELITZ, G. B. 1930 Trans. A.S.M.E., vol. 52, APM–52–5, "Performance of Oil-ring Bearings".

KARELITZ, G. B. 1935 *Mechanical Engineering*, vol. 57, p. 292, "End Leakage in a 120 deg. Bearing".

NEEDS, S. J. 1934 Trans. A.S.M.E., vol. 56, APM–56–16, p. 721, "Effects of Side Leakage in 120 deg. Centrally Supported Journal Bearings".

PEARCE, E. S. 1936 Trans. A.S.M.E., vol. 58, RR–58–1, p. 37, "Locomotive and Car Lubrication".

LUBRICATION OF TEXTILE SPINDLES

By H. G. King *

Types of Spindles. Textile spindles may be divided into three classes: (a) mule, (b) flyer or speed, and (c) ring spindles, but of these the mule and flyer spindles need not be considered, as the speeds at which they run do not in themselves lead to any lubricating problems or difficulties.

Ring spindles are generally of the so-called flexible bath-lubricated pattern. The spindle consists of a bolster or inner tube having a footstep bearing supporting the steel spinning spindle or blade and embodies a bearing surface to resist the pull of the driving band which transmits rotation to the spindle through a small pulley or warve.

The bobbins upon which is wound the yarn, whilst being twisted and spun, upset the balance of the rotating parts and they then easily cause vibration. In the flexible spindle, this is met by allowing the centre of gyration of the unbalanced spindle and bobbin to have sufficient play to assume such a position that steady rotation ensues. This play is allowed for by the inner tube or bolster having a limited amount of play, but is at the same time cushioned by oil, springs, or other means. The pull of the driving band is also made to act near the neck of the inner tube, preferably at such a point that the neck of the inner tube is kept as steady as possible so long as there are no forces which tend to cause vibration of the spindles.

Several commonly used types of spindles are shown in Figs. 1–5. Spindle speeds of between 10,000 and 12,000 r.p.m. are in common use.

Materials Used. The blade is invariably made of steel containing 1 to 1·2 per cent carbon and is heat-treated after forging and grinding. This ensures long life and prevents any likelihood of vibration due to bending being set up under working conditions. The driving sleeves or warves are made of close grained cast iron or aluminium alloy. The bolsters or inner tubes are mostly of cast iron and it is essential that the material be of good quality to withstand wear. The working life of these varies with the load imposed, care in lubrication, and cleanliness, but 10 to 15 years' running is quite common. The collars or bolster carriers are almost invariably of cast iron and must be of such a nature as to be free from porosity, as this part forms the oil bath or well. Built-up and solid-machined steel carriers have been tried, but

* Works Director, The British Northrop Loom Company, Ltd.

are more costly, although they overcome the trouble of porosity where the design makes the walls of the oil well too thin.

Lubrication. The bearing portion of the steel spindle blade is tapered, while the inner tube is bored parallel, so that there is a tapered annular

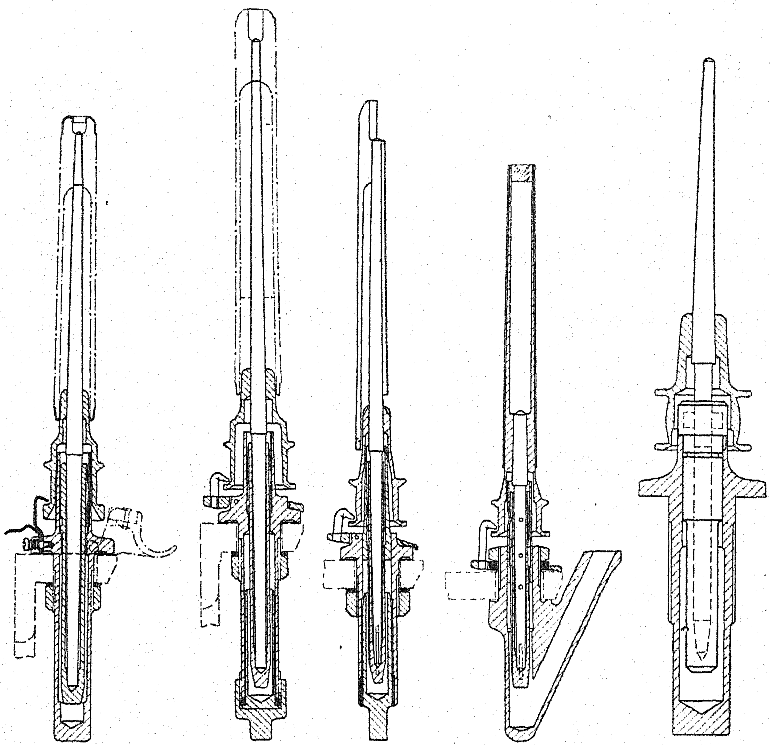

Fig. 1. Solid Bottom or Ordinary Spindle
Fig. 2. Carrier with Brass Cartridge Oil Tube
Fig. 3. Bolster Carrier with Multiple-Screw Oil Tube
Fig. 4. Carrier with Front Oil Spout
Fig. 5. Roller Bearing Spindle

space between the two, thus causing a pumping action on the surrounding film of oil which is forced up into the neck bearing, whence the oil passes over the edge of the inner tube and flows down through flutes applied to the outer neck of the bearing of the bolster and so back into the oil receptacle of the spindle bolster. The oil circulation is assisted by a

number of holes and slots in the lower part of the inner tube. In early days, oiling of the spindles was carried out once a day whilst running at only limited speeds and this was decreased to once a week on the introduction of the flexible spindle, but to-day spindles may run for long periods without attention and at very considerably increased speeds. Oil cups when used (Fig. 3) should not be used for replenishing the oil, but only for cleaning the foot and sediment. If the oil cup is used for filling when screwed into the carrier, the displaced oil will run out down the sides of the cup until it has come to the level displaced by the spindle. In most cases, the oil space of oil cup spindles is smaller than that of the solid type and the oil must therefore be renewed more often. When oiling, the spindle and inner tube or running parts should first be taken out and wiped free from dirty oil and sediment, dried and at the same time inspected for wear and adhering impurities. The old oil is then pumped out of the carrier, sediment removed, fresh oil filled in and the running parts replaced.

Lubricant. For the lubrication of ring or other bath-lubricated spindles, it is necessary to use an oil of the lightest possible body that will maintain a complete lubricating film with minimum friction and lowest operating temperature. The high speeds and rapid circulation of oil through the fine clearances between spindles and inner tubes make it essential that the oil in use shall have maximum fluidity compatible with effective lubrication and be of such a nature that it will not oxidize readily or form gummy deposits with continued use. Oil of which 50 cu. cm. will at 70 deg. F. run through a standard Boverton Redwood viscometer in 68 sec. is suitable. Average oil temperatures under mill conditions may reach 80–100 deg. F.

Wear. Numerous experiments have been carried out to examine both the material and wear of inner tubes. Bronze, lignum vitae, cast iron, alloy cast irons, and fabric resins have been used, but cast iron predominates and, for economy and cost, is difficult to replace as a material suitable for these tubes. Cast iron as generally used is of a close grained homogeneous quality. Most makers anneal this iron, though some use it unannealed, and experiments have been carried out by the author to ascertain which of the two methods is preferable. A small testing machine was made, consisting of a revolving wheel made of spindle steel having the same degree of hardness. This was loaded by predetermined weights and run for a constant period of time at constant speed revolution, and the length of the impression made on the specimen measured. The bolsters of various grades of cast

iron were sectioned longitudinally and run in a bath of oil, thus reproducing the actual running conditions of the spindle.

The phosphorus content influences the resistance to wear, but does not increase it to the same extent as combined carbon. To obtain real wear resistance a pearlitic matrix is essential. The solution is to adjust the phosphorus content and combined carbon content so that the tubes can be machined in the unannealed condition without too great an increase in machining costs. Experiments seem to indicate

TABLE 1. WEAR TESTS OF VARIOUS CAST IRONS

No.	Mixture: Total C	Si	S	P	Mn	Condition	Average wear,* mm.	Average wear in diameter,† inches
Ordinary	3·25–3·60	2·00 to 2·50	0·080 0·100	0·85 1·20	0·50 0·75	Annealed	5·0	0·009
						Unannealed	2·0	0·003
Beam	3·20–3·40	1·70 to 2·00	0·080 0·100	0·65 0·80	0·50 0·60	Annealed	3·2	0·004
						Unannealed	1·75	0·003
Hematite	3·8	2·80	0·036	0·06	0·81	Annealed	7·5	0·021
						Unannealed	3·8	0·011

* Load $9\frac{3}{4}$ lb.; time 15 mins.; lubricated.
† 6 months running tests.

that the best conditions are obtained with 0·50 per cent combined carbon with a network of phosphide eutectic.

Considerations of Design for Lubrication. It is important that the inner tube should have a fairly large space for an oil well at the bottom of the bolster carrier, to ensure that the spindle will run for a long time without requiring re-oiling. The oil chamber should be of a shape which permits any sediment to settle out of the way of the oil circulation, and it must be easy to clean and replenish.

The spindle blade is ground to a true cone having a taper of $\frac{1}{400}$ inch per inch length at the top bearing. The bearing of the inner tube is bored parallel, thus giving a distinctly greater clearance at the bottom of the bearing. The diametral clearance between the bore of the inner tube and spindle at the top should be from 0·0015 to 0·002 inch. Tests indicate that these allowances should not be decreased.

All inner tubes are countersunk at the top of the bore with a 90 deg. countersink to a depth of $\frac{1}{16}$ inch.

The footbearing is made to an angle of 95 deg.; the point of the spindle is always hardened and care taken to ensure that the angle of the footstep bearing in the bolster is not less than that of the spindle itself. If too neat a fit is made at this cone, the spindle tends to jump when running. Some spindles are made with a flat bottom foot resting upon a hardened ball. This type gives good results but lacks the centring effect of the pointed spindle.

At the bottom of the inner tube, to ensure adequate lubrication saw slits are made. These are preferable to the holes which are often used.

Roller Bearing Spindles. The most recent departure during the last 50 years from the ordinary plain bearing type of spindle has been the introduction of ball or roller bearings, for which many advantages are claimed. Fig. 5 represents such a design of roller bearing spindle and for it are claimed:—

(1) Easier starting and higher speeds.
(2) Power consumption smaller.
(3) Lubrication costs less, it being necessary to lubricate only once a year.
(4) Greater cleanliness, because the oil or grease is retained in a perfectly tight housing.
(5) Yarn more uniform because there is less slipping where a free running roller bearing spindle is employed than when the spindle is an ordinary plain bearing spindle.

Most of the claims made may at first sight appear attractive. The greatest difficulty in design is to find room to fit a roller or ball bearing of sufficient size to absorb vibrations due to bad bobbins and out-of-balance loads and these lead to rapid destruction of the races, whilst the greatly increased initial cost renders savings in power consumption and lubricant somewhat doubtful.

THRUST BEARINGS: THEORY, EXPERIMENTAL WORK, AND PERFORMANCE

By Professor Dr.-Ing. Dr. techn. h.c. E. A. Kraft *

Of the three forms of friction—dry, semi-fluid, and completely fluid friction—the last-named is always aimed at for thrust bearings as well as for journal bearings of high circumferential velocity. Experience with modern thrust bearings proves that the now usual method of calculation is sufficiently accurate.

Formerly the problems presented by thrust bearings received but imperfect solutions. Particularly owing to confusion about methods of lubrication, limiting values had to be observed, on experimental and observational grounds, of 3–8 kg. per sq. cm. for the bearing pressure and a maximum sliding velocity of 10 metres per sec. on the average segment. For this reason, a single thrust collar could only take up a comparatively small thrust; as a remedy, series of thrust collars were arranged one behind another on a shaft. Apart from the fact that such a multiple-collar thrust bearing must have large dimensions in order to take up large thrusts, owing to the inevitably unequal heating of the rotating and stationary parts, which are mostly made of different materials, it is almost impossible in practice, even with the most careful manufacture and supervision, to arrange that all the thrust collars shall take up equal loads. As this method of construction leads to difficulties both in installation and in lubrication, it is obvious why multiple-thrust collar bearings have enjoyed no great popularity.

Modern theory (Reynolds, Gümbel, Michell) lays down two especially important conditions for thrust bearings. First, the pressure in the liquid layer between a moving and a stationary surface can only rise in the direction of flow, i.e. the direction of one of the moving surfaces, when the viscosity η of the liquid is not equal to zero. A frictionless liquid is thus unsuitable to take any thrust.

Similarly, the pressure in the liquid layer can only rise when the clearance between the two sliding surfaces, and therefore the thickness of the oil layer, is variable. On the other hand, when the two opposed surfaces are parallel, they are unsuitable for taking pressure unless oil is introduced at a higher pressure than the surface pressure.

These conditions led Michell to develop his thrust bearing in which the stationary surface is split into individual pads which take up an oblique position in respect to the collar under the influence of velocity and load. Between the collar and the stationary pressure surface,

* Allgemeine Elektricitäts Gesellschaft, Berlin.

several liquid wedges are formed which are able to resist extremely high pressures. An example of this thrust bearing designed for heavy load is given in Fig. 1.

Relations of Pressure and Friction. With a surface of length l measured in the direction of movement and breadth b (vertically to l) it is possible to calculate, without any special difficulty, the pressure distribution over the surface, the highest surface pressure, the total pressure and its point of application, together with the coefficient of friction μ. Thus for the pressure distribution one obtains the well-known pressure curve with the highest pressure as well as the point of

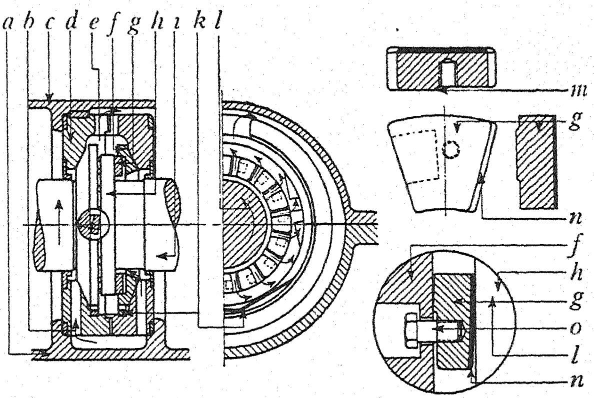

Fig. 1. Single-Collar Thrust Bearing with Movable Pads for Thsusts in Both Directions on a Horizontal High-Speed Shaft

a	Pedestal.	*h*	Thrust collar.
b	Oil inlet.	*i*	Shaft.
c	Bearing cover.	*k*	Supporting spring.
d	Thrust bearing housing.	*l*	Direction of rotation.
e	Oil drainage.	*m*	Tilting edge.
f	Supporting ring.	*n*	Oil inlet edge of pad.
g	Movable pad.	*o*	Screw.

application of the total load behind the centre, in the direction of movement. The coefficient μ depends on the relation l/b and even on l, as μ increases with increasing ratio l/b, while at the same time the point of application of the resulting pressure moves to a small extent towards the exit edge of the pad. For values of l/b between 0 and 1·0 the coefficient of friction remains approximately unchanged and can only be expressed numerically for definite conditions. If the figures for pad thrust bearings are compared with those for thrust bearings with undivided pressure surfaces, it is seen that owing to the subdivision

of the surface and the formation of a wedge of oil, μ decreases to roughly $\frac{1}{10}$ to $\frac{1}{20}$ of its former value (about 0·03), assuming values of approximately 0·003 to 0·0015.

The Pressure Pad. The formation of a supporting oil film depends essentially on the form of the individual pads, which should not be too long, measured in the direction of rotation, and must be broad in the perpendicular direction since otherwise most of the oil will escape at the edges. Further, it is essential that the point of tilting or the tilting edge must be arranged so that the pressure surface sets up a liquid pressure so as to form an oblique wedge of oil. The pad must thus be supported, looking in the direction of rotation, behind the point of incidence of the resulting liquid pressure. Opponents of the single-collar pad thrust bearing have challenged its introduction on the theoretical ground that the exact position of the support must change for every load. However, to-day, thousands of thrust bearings of various dimensions are being used and no trouble has been experienced from any slight unavoidable inaccuracy in manufacture. Experience has shown that the self-regulation of the pads goes further than theory indicates and that the laws deduced theoretically apply fundamentally for bearings which only approximately correspond to the strict values. Theory indicates that the position of the support of the pressure surface for the least friction is about $0{\cdot}4l$ from the exit edge.

Lubricants and Bearing Metal. The higher the viscosity of the lubricant, the higher the loads that can be borne by a pressure pad. An oil utilized for high velocities ("Turbinenöl") has a flash point of 180 deg. C., and a viscosity of 12–20 deg. Engler at 20 deg. C., and 2·5–4 deg. Engler at 50 deg. C. The quantity of lubricant required for a single-collar pad thrust bearing, allowing for a temperature rise of 5–10 deg. C. over an inlet temperature of 45–50 deg. C., depends chiefly upon adequate removal of heat. The relation of p and v, perhaps in the form $\sqrt{pv^3}$, is the same as for journal bearings. The cross-section of the oil inlet within the bearing should be generously proportioned, and the liquid velocity should not exceed 1·25 metres per sec. at entry and 0·75 metre per sec. at discharge.

Both self-lubrication and positive force-feed lubrication are used. Self-lubrication is often used where loads are small, though large, slow-running bearings, particularly bearings for water turbines and marine shaft thrust bearings, are often self-lubricated. If forced-feed lubrication (pressure about 0·5 atmos.) is to be utilized in, say, a steam turbine, it will also be used for the thrust bearing. The oil should

always be fed to the shaft on the inner edge of the bearing surface and leave at the highest point of the housing, so that, when running, the whole bearing is always full of oil.

Pressure pads are made of steel or bronze, with whitemetal linings on the bearing surface. Usually the percentage composition of the whitemetal is: tin, 80; antimony, 12; copper, 6; lead, 2; *or* tin, 10; antimony, 16; copper, 1; lead, 73. Given abundant lubrication with good oil and a load within permissible limits, the choice of bearing metal is immaterial, though choice becomes important at the onset of semi-fluid or dry friction. The thickness of the whitemetal layer will generally be less than the unobjectionable yet permissible axial displacement of the shaft so that when, owing to some defect the whitemetal has to yield or run, serious damage to the machine is avoided. In itself the thickness is immaterial, but great care is required in preparing thin whitemetal linings owing to the difference in thermal expansion between whitemetal and bronze or steel and to their unequal rates of cooling.

Computations. Apart from the fundamental researches of Michell, only isolated investigations have been carried out. Thus Messrs. Brown, Boveri and Company have investigated square pressure pads under loads ranging from 0–100 kg. per sq. cm., velocities from 13–36 metres per sec., and temperatures from 70–100 deg. C.* Amongst the earliest researches in which complete thrust bearings made for definite objects and conditions were tested for their load capacity and reliability, were those of the American Westinghouse Company on a single-collar pad bearing. At an average velocity of roughly 17 metres per sec. the experimental bearing remained perfectly reliable up to a load of 70 kg. per sq. cm., though at a surface load of roughly 500 kg. per sq. cm., there was no appreciable damage. The bearing became unserviceable only at a load of 750 kg. per sq. cm., when the whitemetal ran out. Extensive researches were also carried out by the Allgemeine Elektricitäts Gesellschaft with single-collar pad thrust bearings at various loads and velocities. With marine shaft bearings moving at the low velocity of 2·4 metres per sec. and a surface load of about 30 kg. per sq. cm. the coefficient of friction μ was 0·0025. The specific pressure could be increased here also to several hundred kilograms per square centimetre of bearing surface.

In choosing dimensions for actual designs, the surface pressure should be fixed at a lower figure. The average load for large pad thrust bearings is usually fixed at 30 kg. per sq. cm., the maximum value for smaller bearings being 20–25 kg. per sq. cm. The highest

* See Brown, Boveri Company, Mitteilungen, 1918, vol. 5, pp. 9, 27, 46, 71.

velocity at the middle diameter of the pads can be fixed safely at 65 metres per sec. As the increase in velocity and load leads to an increased production of heat, therefore, when one of the two values becomes disproportionately high, the other must be correspondingly decreased. In general it is advisable to increase the load to the upper limit if possible, to economize in respect of circumferential velocity and external dimensions.

Examples. Thrust bearings with complete pressure disks are occasionally utilized for small thrusts, often combined with a pad thrust bearing in such a way that the pad thrust bearing takes up the main thrust, while the bearing with the undivided bearing ring only has to take up thrust in the reverse direction. These bearings have surfaces with a fixed angle of inclination, as the bearing surface is split up into separate portions by radial grooves, the corresponding edges being rounded off and sloped (Fig. 5) on the left side of the collar.

In designing a single-collar pad thrust bearing two main points should be remembered: (1) each individual pad must take up the desired inclination with ease; and (2) all the pads must bear equally. Both requirements are met in various ways in practice. While the former depends on the construction of the pads, the latter is influenced by the shape of the supporting members and plates. For simple bearings under slight thrust and for journal bearings placed at small distances apart, a flat supporting plate will suffice. If a bearing without any special adjusting device must place itself on shifting or sagging shafts so that the bearing surface always remains truly perpendicular to the axis of the shaft, then the supporting ring must incorporate a spherical seating or the whole thrust bearing must be combined with a journal bearing supported on a spherical surface (Fig. 5).

The tilting movement of the individual pads must be as free as possible. An ideal method of support would be point contact between pad and support, which would also permit the pads, e.g. through elastic deflexion of the collar, to adjust themselves to the obliquely placed collar by taking up a radial inclination. This has the disadvantage that a very small bearing surface has to meet the whole thrust of the pad; this, like the abutment, must be made of very hard material or this type of support will only be suitable for comparatively small thrusts.

In the small turbine thrust bearing shown in Fig. 2, the pads are supported by pins which have a hardened ground plane surface and are seated in the convex recess of the pad. The pins are made of steel and the pads of bronze covered with whitemetal. The collar is one

with the shaft. The thrust bearing can take up axial thrust up to 1 metric ton at 7,000 r.p.m. Both the journal and the thrust bearing have one common lubrication system. As the machines for which this bearing is intended have only a small bearing space, a spherical support is avoided here. Bearings of this kind have given very good results. The great advantage is that they are simple to manufacture and assemble. The auxiliary bearing for thrust opposed to the main direction is placed on the other end of the adjacent journal bearing and is a simple collar bearing.

The pads of a water turbine bearing (Fig. 3)* also have approxi-

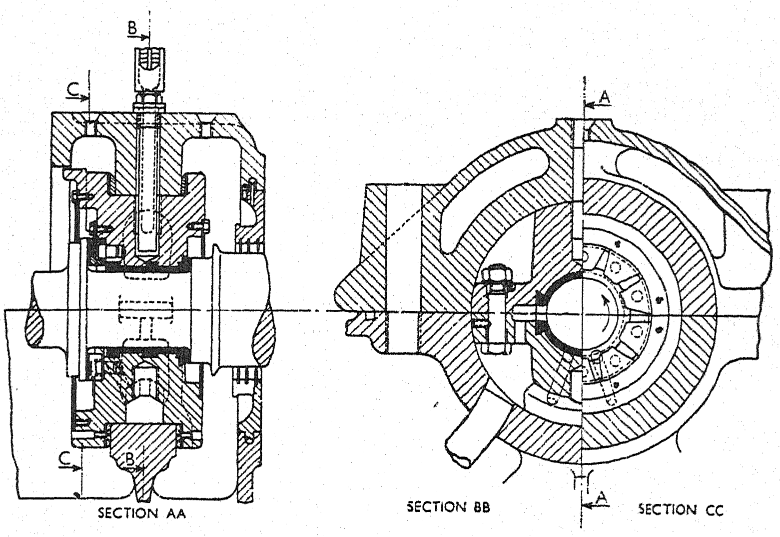

Fig. 2. Pad Thrust Bearing for Slight Thrusts and High Shaft Speeds for a Steam Turbine (Allgemeine Elektricitäts Gesellschaft)

mately a tilting point arrangement. The pads end in steel pins with a spherical under-surface, which lies on a soft iron collar, into which it is pressed when under load. Small mechanical inaccuracies are compensated by the more or less strong pressure of the spherical seats, and are thus rendered harmless. Each of the ten pads takes up a load of roughly 90 metric tons, the external diameter of the pad collar is 2,300 mm., and the speed of the turbine 75 r.p.m.

In another type of spherical support for steam turbines (Fig. 4) each pad rests with its oblique surface on two balls, so that one common ball touches two adjacent pads on facing edges. This ensures

* Escher Wyss Mitteilungen, 1936, p. 64, Fig. 8.

an equal distribution of the load, as each more heavily loaded pad on its spherical support presses its neighbouring pad the more strongly

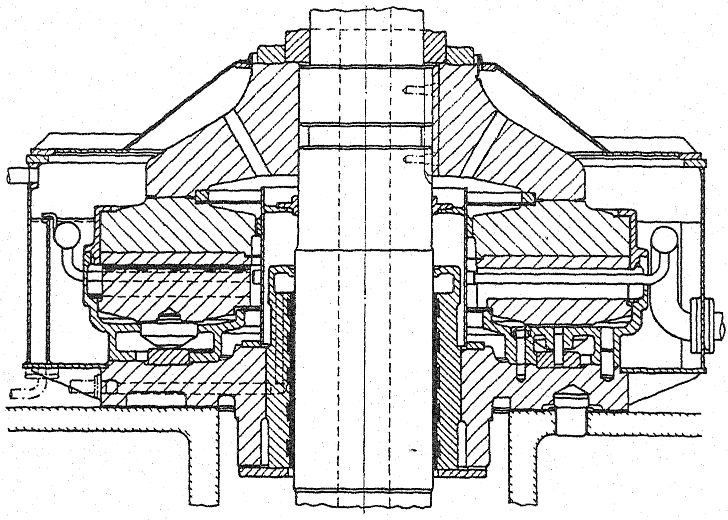

Fig. 3. Single-Collar Pad Thrust Bearing for a Water Turbine (Escher Wyss and Company)

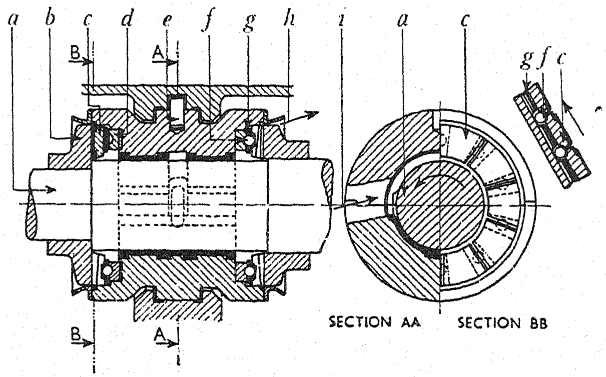

Fig. 4. Pad Thrust Bearing for Equal Thrust in Both Directions for a Steam Turbine (Brown, Boveri and Company)

a	Shaft.	*f*	Ball.
b	Collar.	*g*	Ball cage.
c	Pad.	*h*	Oil drainage.
d	Supporting ring.	*i*	Oil inlet.
e	Journal bearing.		

against the thrust of the collar until the pressure is equally distributed. The tilting movement is also favoured in this way because the pads have different obliquities. This bearing is also combined with the journal bearing, and is constructed to take up large thrusts in both directions. To weaken the action of the very hard balls upon the somewhat softer pads, steel plates are placed at places of contact. A spherical arrangement of supporting plates can be omitted in bearings of this kind, as the ball-locking chain adjusts itself with equal load distribution on the somewhat inclined surface of the collar.

While these examples only deal with the point support of the pads, the following bearing is characterized by line support. The pads tilt on a tilting edge, which is naturally in a radial position and, like the

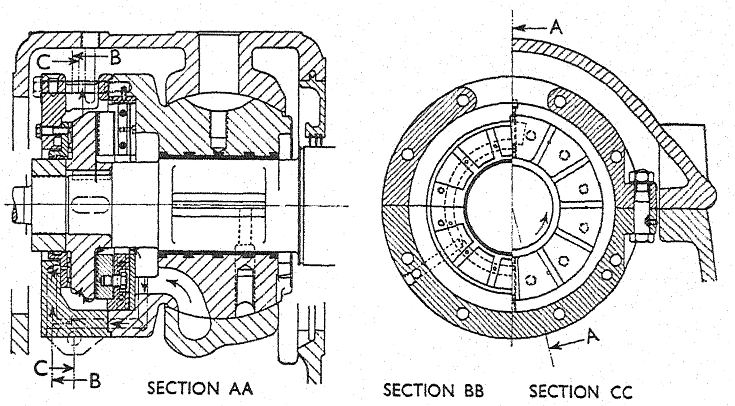

Fig. 5. Pad Thrust Bearing for Mainly Unilateral Thrust for a Steam Turbine (Allgemeine Elektricitäts Gesellschaft)

supporting point, must be behind the middle line of the pad. The steam turbine bearing illustrated (Fig. 5) is for a predominantly unilateral thrust; for a thrust in the opposite direction only a rigid running surface is provided. As the thrust bearing is combined with the adjacent spherically supported journal bearing, it can adapt itself to an oblique position of the collar owing to the bending of the shaft, without supporting the pad ring through spherical surfaces. The bearing is built for a thrust of 8 tons in the main direction at 3,000 r.p.m.

Other balancing arrangements have been used to bring independent elastic members directly between the pads and the supporting ring or bearing body. One such proposition, due to Kingsbury, is to provide single pads with spiral springs housed in a bore which act as a kind of oil-buffer.

The main application of single-collar pad thrust-bearings is in marine propelling shafts, and in steam and water turbines; and it is being extended to revolving prime movers, pumps, and centrifugal compressors. To appreciate fully the importance of these innovations in thrust bearings, it is often necessary to know the influence of the thrust bearing on the design of the machine itself and its duties, a subject which can only be mentioned in passing. Recent steam turbines with their high efficiency, compact construction, and small weight, can hardly be imagined without the single-collar pad thrust bearing. In the same way it has considerably extended, both in simplicity and reliability, the constructional possibilities of water turbines, electric generators, and marine engines.

THE TAPERED-LAND THRUST BEARING

By F. C. Linn and R. Sheppard*

The tapered-land thrust bearing which has fixed converging surfaces such that a wedge of oil will exist between the rotating collar and the thrust plate, was developed for the first 10,000 kW. 3,600 r.p.m. turbine-generator set built by the General Electric Company, and has since been applied on all types of turbine units and is now the company's standard. The theory of using converging surfaces between a moving and stationary member to produce and maintain an oil film which will support heavy loads was originally presented by Osborne Reynolds and has long been established. Thrust bearings with fixed tapers have been used by S. Z. de Ferranti and others from time to time. The success with the taper-land thrust bearing may be largely attributed to the uniform accuracy produced by special tools used in machining the lands. The thrust load of a horizontal steam turbine is negligible at zero speed and increases from a small value at zero output to a maximum at or near maximum output at rated speed. This condition is ideal for the adaptation of the tapered-land thrust bearing since the bearing can be designed with proper dimension and taper to carry the imposed load.

The tapered-land thrust bearing (Fig. 1) consists of a collar A rigidly attached to the shaft and two stationary Babbitted plates. Ordinarily, one of the plates is the loaded plate B and is machined with tapered lands, while the other plate C with flat lands limits the end play. The housing for the thrust plates is bolted to the side of the shaft journal bearing which is supported by a ball-seated mounting. In other cases, the housing for the thrust plates is separately mounted in a ball seat of its own. The collar has two continuous working faces which have no radial grooves. The loaded side usually has a greater diameter than does the unloaded side. The diameter of both sides is the same if a turbine has a possibility of two-directional thrusts (as in double-flow turbines).

The loaded thrust plate has a split steel ring with a Babbitted face divided into lands by radial oil-feed grooves. The surface of each land is tapered (Fig. 2), so that it slopes relatively to and towards the rotating collar both in the direction of rotation and from the inner to the outer radii at the leading edge of the land. A suitable approach curvature from the oil feed groove to the tapered surfaces is provided at the leading edge of the lands. Fig. 2 shows the constructional

* General Electric Company, Schenectady, U.S.A.

details of one of the lands of the loaded plate. The land is bounded by the four corners, A, B, C, and D. The points A and B at the trailing edge of the land are in a plane parallel to the collar; the point C is in a plane below points A and B and the point D is in a plane below C. The dimension Q of sections RS and OP is of that portion of the land

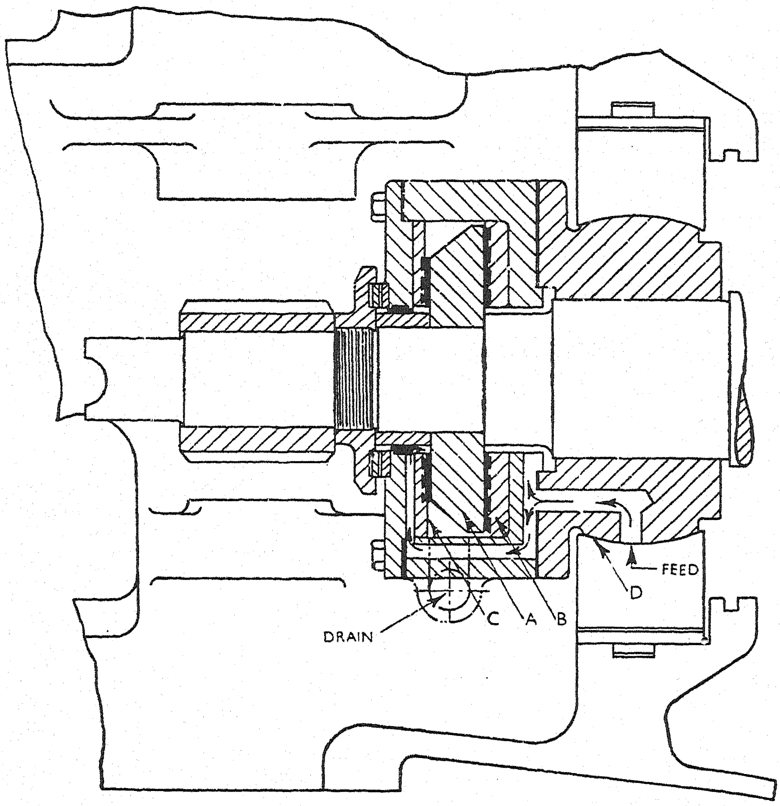

Fig. 1. Tapered-Land Thrust Bearing

which is made parallel to the surface of the collar and is very small on a steam turbine that rotates only in one direction.

On reversing marine turbines, the flat portion of the land may occupy as much as one-half the total land area, and suitable approach curvature from the oil feed grooves is provided at both ends of the lands. The direction of thrust load may change with the reversed rotation, so both plates are made with tapered lands with a relatively large proportion of the land parallel to the runner. The tapers of the lands on the two

plates are cut in opposite directions so as to have correct relationship to the runner during their respective directions of rotation.

Constructionally, the unloaded thrust plate is generally the same as the loaded thrust plate, except that the surfaces of the lands are parallel to the surface of the rotating collar. A suitable approach curvature from the oil-feed grooves to the surfaces of the lands is provided.

Cooled oil is admitted under pressure to the radial oil-feed grooves in the thrust plates, whence it is carried over the surfaces of the lands

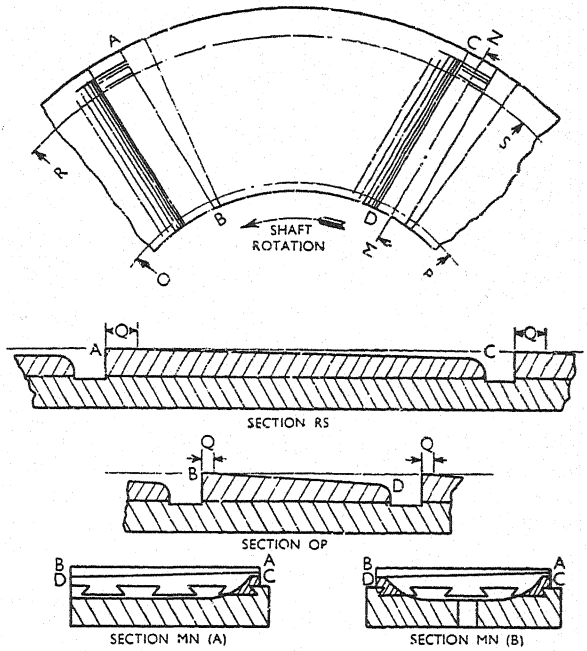

Fig. 2. Face and Sectional Views of Tapered Land and Oil Grooves

by the rotating collar, thus forming a continuous film of oil between the collar and plates. The radial oil-feed grooves are dammed at the outer ends when oil is fed from the inner radius (Fig. 1 and Fig. 2, section MN (A)), and at both ends when oil is fed to the central portion of the grooves (Fig. 2, section MN (B)) so as to maintain a pressure in the groove above atmospheric pressure. The oil, after leaving the thrust surfaces, is returned to the tank.

The tapered-land thrust bearing is simple in construction, and when properly designed has high load-carrying capacity and low power loss.

Tests show that loads of 1,000 lb. per sq. in. can easily be carried, so they are being applied for working loads from 250 to 500 lb. per sq. in., thus having an ample factor of safety.

As previously stated, the tapered-land thrust bearing has been successful mainly because of the tools used in the manufacture of the plate with the tapered lands. After the two surfaces of the plate have been machined parallel and the radial grooves milled, the plate is placed on a special machining fixture with which the tapered surfaces are duplicated and accurately machined.

Design of Tapered Surfaces. The proper taper to give the desired load-carrying capacity at a given speed, absolute oil viscosity, and film thickness is determined by formulæ based on Gümbel's theory. The minimum film thickness to be used in design has been determined by test. Tests indicate that the minimum design clearance should not be the same for all thrust bearings but should be greater on large than on small bearings. A minimum clearance of 0·5 mil (1 mil=0·001 inch) is satisfactory on the small, high-speed (15,000 r.p.m.) bearings, whereas a minimum clearance of 1·5 mils is necessary on larger bearings operated at lower speeds (1,500 r.p.m.).

The calculated taper is for the mean radius of the lands. Best results are obtained when the taper, in mils, is least at the outer circumference and greatest at the inner circumference. The design of the taper of the lands is, therefore, the calculated value at the mean radius, a lesser amount at the outer radius, and a greater amount at the inner radius, and the amount of change from the calculated value is based on judgement.

Tests. Tests have been conducted on thrust bearings designed for 20,000 r.p.m., at 3,600 and 1,800 r.p.m. The object was to determine the load-carrying capacity, power loss, and the relative merits of the bearings tested. The design of all the test stands was similar (Fig. 3). The shaft on which the thrust collars were mounted was supported by journal bearings and driven through a flexible coupling. One of the thrust bearings was mounted in the housing. The other was centred by a rabbet and bolted to a stub shaft coupled to a piston or diaphragm. Hydraulic pressure was impressed on the piston or diaphragm to the desired amount, the force of which was directly transmitted to the thrust bearings being tested. Calibrated pressure gauges, oil flow meters, and Fahrenheit thermometers gave accurate readings for determination of the thrust load and power loss.

The results of tests made at 20,000 r.p.m. on a six-land thrust bearing $1\frac{1}{8}$ inches in internal diameter by $1\frac{7}{8}$ inches external diameter are shown

in Fig. 4. It is interesting to note that a plate having lands with zero taper had a greater load-carrying capacity and less power loss than a plate with lands with too great a taper. However, a plate with properly

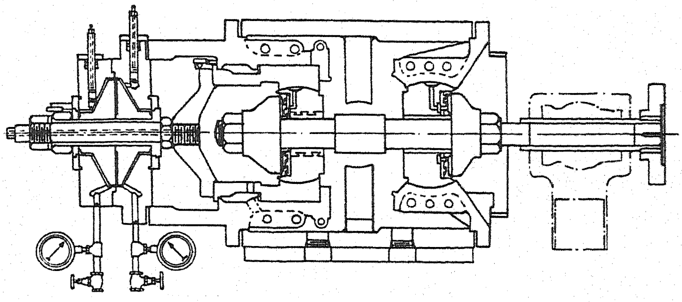

Fig. 3. Machine for Testing Thrust Bearings

tapered lands carried a load up to the capacity of the test stand without failure.

The load–power loss curves of tapered-land thrust bearings tested at 3,600 r.p.m. (Fig. 5) show that a plate with lands of zero taper has much less load-carrying capacity than has one with tapered-land plates.

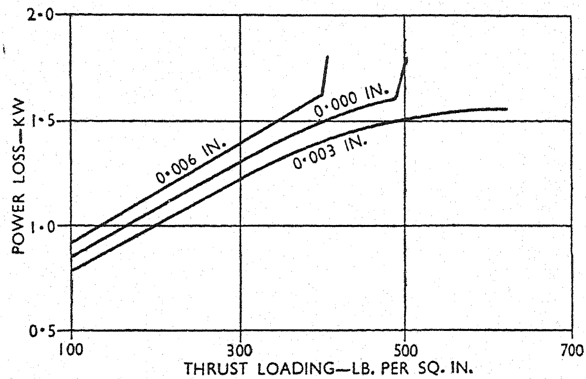

Fig. 4. Power Loss as a Function of Load at 20,000 r.p.m.

Viscosity of oil, 280 sec. Saybolt at 100 deg. F. and 49·5 sec. Saybolt at 212 deg. F. Inlet oil, 120 deg. F. Net area of bearing 1·44 sq. in.

The plates with the largest tapers have greater loss and less load-carrying capacity than the plates with lesser tapers; and, finally, a plate with a greater taper at the inner than at the outer periphery has the least loss and greatest load-carrying capacity. These tests indicated

that increasing the amount of the taper at the inner periphery over that of the outer periphery permits a larger percentage of the oil to be carried on to that portion of the land which feeds the trailing edge, resulting in a thicker oil film, cooler operation at the higher loads, and higher load-carrying capacity.

On testing the thrust bearing at 1,800 r.p.m. (Fig. 6), a load of 1,100 lb. per sq. in. was carried without failure. The temperature of

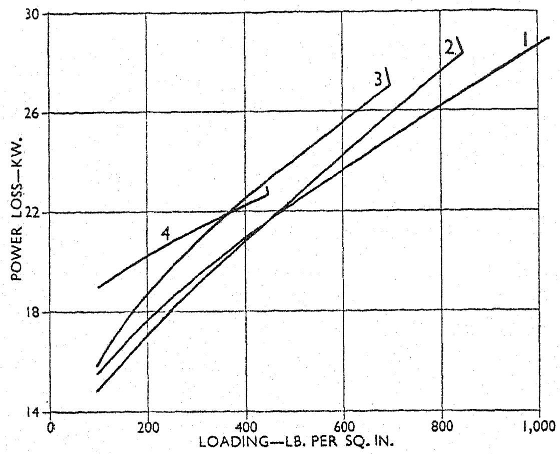

Fig. 5. Power Loss as a Function of Load at 3,600 r.p.m.

Viscosity of oil, 150 sec. Saybolt at 100 deg. F., 43 sec. Saybolt at 212 deg. F. Inlet oil temperature, 120 deg. F. Oil flow, 11 gal. per min. Oil pressure in radial feed grooves, 2–3 lb. per sq. in. Loss for loaded plate only (outer diameter, 11½ inches; inner diameter, 6 inches; effective area, 60·5 sq. in.).

Lands	Taper at Outer diameter, inch	Taper at Inner diameter, inch
8	0·006	0·009
8	0·006	0·006
10	0·010	0·010
10	0	0

the Babbitt metal at the trailing edge increased with the power loss and with an increase in load.

Applications. As a result of these tests, and experience with units operating at 3,600 r.p.m. with thrust loads up to 450 lb. per sq. in., a line of thrust bearings has been developed for turbine sets, including marine installations, for speeds varying from 1,500 to 20,000 r.p.m., ratings varying from 100 kW. to 160,000 kW., and for maximum thrust loadings from 150 to 450 lb. per sq. in. Several hundred units

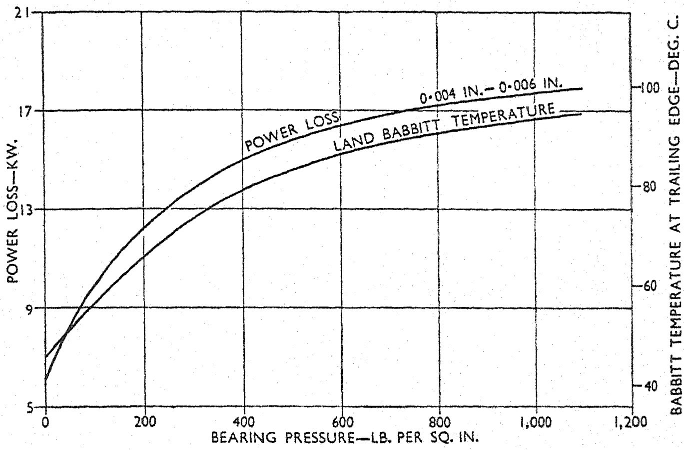

Fig. 6. Power Loss as a Function of Load at 1,800 r.p.m., Including the Temperature of the Land Babbitt Face

Viscosity of oil, 375 sec. Saybolt at 100 deg. F., 51 sec. Saybolt at 210 deg. F. Inlet oil temperature, 113 deg. F. Power loss for loaded plate only (outer diameter, 14 inches; inner diameter, 5·5 inches; effective area, 111 sq. in.). Six lands: outer diameter, 0·004 inch taper; inner diameter, 0·006 inch taper.

with tapered-land thrust bearings installed are in service; and future units, to the highest rating, will be equipped with this bearing. The broad range of conditions for which a line of tapered land bearings has hus far been designed and operated is illustrated by Table 1.

TABLE 1. DIMENSIONS OF TAPERED-LAND THRUST BEARINGS, AND THEIR APPLICATIONS

Net area, sq. in.	External diameter, inches	Internal diameter, inches	Number of lands	Rating, kW.	Speed range, r.p.m.
1·44	$1\frac{7}{8}$	$1\frac{1}{8}$	6	50	20,000
12	5	$2\frac{3}{4}$	6	200–2,500	3,600–14,000
24	$7\frac{1}{2}$	$4\frac{1}{2}$	6	1,000–4,000	3,600–8,000
35	9	$5\frac{1}{2}$	8	2,500–6,000	3,600–5,000
68	$11\frac{1}{2}$	$5\frac{1}{2}$	8	5,000–15,000	3,600
90	$13\frac{1}{2}$	$6\frac{5}{8}$	8	2,500–40,000	3,600
106	$13\frac{3}{4}$	$5\frac{1}{2}$	6	40,000–60,000	1,800–3,600
120	$14\frac{3}{4}$	$5\frac{3}{8}$	8	110,000	1,800
165	$18\frac{1}{4}$	$9\frac{1}{2}$	8	35,000–160,000	1,500–1,800
180	$21\frac{3}{4}$	14	12	75,000	1,800
275	$24\frac{3}{4}$	14	12	65,000	1,800

Conclusions. The tapered-land thrust bearing has come into commercial use because of its simplicity, ability to withstand high unit loading, low power loss, and because of the success it has experienced in a wide variety of applications.

The authors wish to express their appreciation of the assistance of their associates and particularly the late Mr. V. Petrovsky, who developed the design formulæ upon which these bearings are based.

JOURNAL BEARING DESIGN AS RELATED TO MAXIMUM LOADS, SPEEDS, AND OPERATING TEMPERATURES.*

By S. A. McKee †

The problem of bearing design is complicated by the fact that a journal bearing is not only required to support its load under all conditions of operation, but also must often provide means for dissipating the heat generated in shearing the film of lubricant. With bearings operating at moderate loads and speeds, the heat generated in the bearing is usually small in comparison with the capacity of the bearing for heat dissipation, and the prime consideration is to ensure that the bearing shall operate with a complete film of lubricant between the surfaces.

Rational methods (Hersey 1915, Karelitz 1930) for the design of bearings operating under constant moderate loads and speeds have been published and need not be discussed here. Two methods for assuring conditions for "thick-film" or stable lubrication have been suggested. One is based upon the evaluation of the minimum thickness of the oil film by means of theoretical hydrodynamical relations involving the bearing dimensions and the assumed operating conditions. The other is based upon dimensional analysis and involves the choice of a suitable value for the generalized operating variable ZN/P. This variable is of particular significance in that it determines the value of both the coefficient of friction and the relative film thickness for a given bearing whenever the bearing is operating in the region of stable lubrication. By its use experimental data may be so correlated as to be directly applicable to design. It is usual to assume a reasonable operating temperature as a basis for a first computation and by methods of successive approximations based on the relations between the heat generated, viscosity, and temperature rise, to determine the operating temperature of the bearing. Recently, however, Hersey (1936) has suggested a three-dimensional graphical method for the simultaneous solution of the equations involving the three variables. The final computation is accurate only to the extent that the relations between the three variables are known.

In numerous applications the modern trend towards higher speeds and loads has reached the point where the factors affecting bearing

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† Mechanical Engineer, National Bureau of Standards, Washington, U.S.A.

temperature are as important as those relating to proper film formation. Many applications also involve wide variations of load and speed. In such cases the designer possibly is more interested in the extreme conditions of load and speed at which a bearing will operate successfully than he is in its performance at normal running conditions. From this standpoint a rational basis for design would seem to be the determination of the maximum allowable loads and speeds predicated upon two primary considerations: (1) that the bearing shall always operate in the region of stable lubrication, and (2) that the operating temperature of the bearing shall never exceed some fixed value. A bearing operating in the stable region will tend to reach equilibrium at some temperature where the rate of heat generation is equal to the rate of heat dissipation. Hence, if the equilibrium operating temperature is to be used as a basis for design, a knowledge of the factors affecting both the rate of heat generation and the rate of heat dissipation is essential.

Heat Generation. Using the nomenclature given in the list of symbols (p. 185), the rate of heat generation may be expressed as

$$H = k_1 F \pi D N \qquad (1)$$

which may also be written as

$$H = k_1 \pi f P L D^2 N \qquad (2)$$

Values for f for a given bearing at various operating conditions may be computed from theory or measured directly. In one investigation (McKee and McKee 1929) the effects of changes in the length-diameter and clearance-diameter ratios were determined over a wide range of operating conditions. The results indicated that an equation of the form

$$f = k_2 \left(\frac{ZN}{P}\right)\left(\frac{D}{C}\right) + \Delta f \qquad (3)$$

is reasonably accurate for the determination of friction losses in small 360-deg. bearings when operating in the stable region. In this equation k_2 is equal to 473×10^{-10} when the units given in the list of symbols are employed. Δf is a correction for changes in the length-diameter ratio. The values to be used for various L/D ratios are shown in Fig. 1.

Substituting equation (3) in equation (2) yields

$$H = k_1 PND \left[k_2 \left(\frac{ZN}{P}\right)\left(\frac{D}{C}\right) + \Delta f \right] \pi LD \qquad (4)$$

Heat Dissipated. The available experimental data on the heat dissipation of bearings are insufficient to provide a rigorous evaluation

of the factors involved. There is some indication (Hersey 1936, Barnard 1932), however, that with certain types of self-cooled bearings the rate of heat dissipation may be approximately represented as

$$H'=k_3A(\Delta T)^{1\cdot 3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

The value for k_3 will depend upon many factors, including the convection currents present. Some data indicate that when the bearing is in a strong draught the value of k_3 may be three or four times as great as when the bearing is in still air.

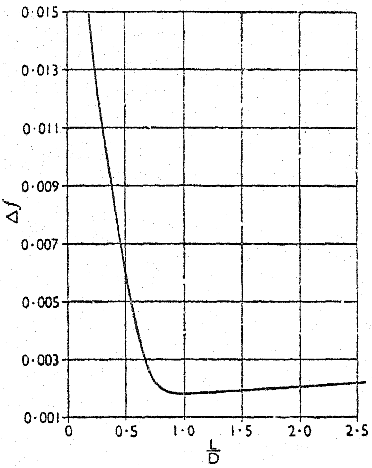

Fig. 1. Values of Δf for Various L/D Ratios: Small-Bore, 360 deg. Bearings

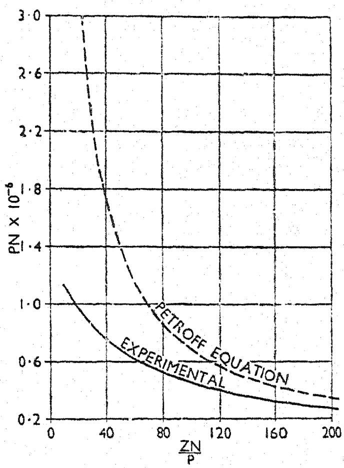

Fig. 2. Effect of ZN/P Value on P–N Relation for a 3 × 3-inch Bearing under Assumed Limiting Conditions

Equation (5) may also be written

$$H'=k_3k_4\pi LD(\Delta T)^{1\cdot 3} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

Since for equilibrium operating conditions, the rate of heat dissipation must equal the rate of heat generation, equations (4) and (6) may be combined to form

$$k_1PND\left[k_2\left(\frac{ZN}{P}\right)\left(\frac{D}{C}\right)+\Delta f\right]=k_3k_4(\Delta T)^{1\cdot 3} \quad . \quad . \quad . \quad . \quad (7)$$

This equation indicates approximately the relation between the rise in temperature of a bearing above its surroundings and the other factors involved, when it is operating in the region of stable lubrication.

Application to Design. Equation (7) involves both the generalized operating variable ZN/P, upon which depends the relative film thickness, and the rise in temperature ΔT which determines the operating temperature of the bearing. This suggests the possibility of obtaining a basis for design by substituting in equation (7) a minimum limiting value of ZN/P to ensure stable lubrication and a maximum limiting value of ΔT based upon a maximum allowable bearing temperature and a maximum temperature of the surroundings. The limiting value for ZN/P could be chosen with a reasonable margin of safety above the minimum point of curves of f plotted against ZN/P where friction data for similar bearings are available, or it may be based upon operating experience. The limiting operating temperature would depend upon the particular application. In some cases the question of oil stability may be the deciding factor while in others the effect of temperature upon the bearing metal may be of more importance. If such limiting values are substituted then for a given bearing, equation (7) will assume the form

$$PN=K \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

wherein K represents the product of the maximum allowable pressures and speeds at which the bearing will operate under the conditions prescribed for safety This equation does not depend upon the particular form of equations (4) and (6), but is applicable to all types of journal bearings where the heat dissipation is independent of the speed of the journal. In some bearings the heat dissipation coefficient may depend upon the speed of rotation. In these cases the form of equation (8), more nearly representative of actual conditions, might be $PN^c = K$, where c is less than 1.

Equation (8) may be written also as

$$PV=K' \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

This is a more general relation where a given value of K′ is applicable to all bearings having the same C/D ratios, L/D ratios, and heat dissipation characteristics. Equation (9) is similar in form to a relation that has been much used as a basis for bearing design, but it should be noted, however, that equation (9) is not a "blanket" relationship that is applicable to all conditions. It involves a definite relation between load, speed, viscosity of the oil, and operating temperature as prescribed by the limits assumed for safe operation.

Numerical Example. To illustrate the application of equation (8) a numerical example is given in which the following assumptions are made: In a 360 deg. bearing with D=3 inches, L=3 inches, C=0·004 inch, conditions for heat dissipation are such that $k_3k_4=0{\cdot}00116$,

minimum allowable $ZN/P=30$, maximum temperature of surroundings $=100$ deg. F., and maximum allowable bearing temperature$=250$ deg. F. Under these assumptions $\Delta T=150$, $D/C=750$, and $L/D=1$, hence $\Delta f=0{\cdot}0018$ (from Fig. 1). By substituting the above values in equation (7) and solving for PN, a value of 850,000 is obtained, which is the value of K in equation (8). Thus with this bearing under the conditions assumed to represent the limits for safe operation the product of the pressure and the speed should not exceed 850,000, or if expressed as in equation (9), PV should not exceed 667,000.

Values for the individual variables P and N when using different lubricants may be obtained by the simultaneous solution of the equations $PN=850{,}000$ and $ZN/P=30$ for a number of values of Z. The results of these computations are given by the P–Z and N–Z

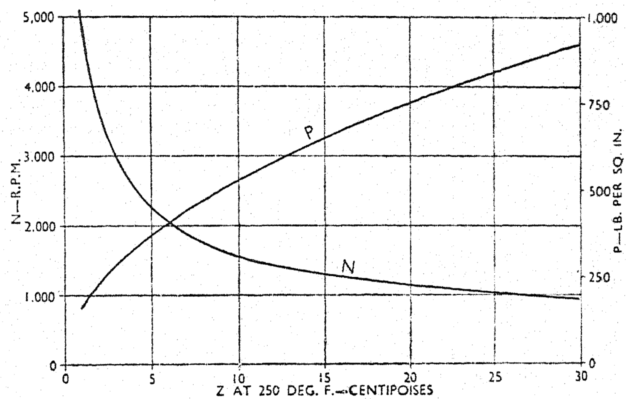

Fig. 3. Pressure-Viscosity and Speed-Viscosity Curves for 3×3-inch Bearing Under Assumed Limiting Conditions

curves in Fig. 3. A point on the P–Z curve represents the maximum pressure at which the bearing may be operated under the given limitations when using an oil having a viscosity at 250 deg. F. as indicated. The point on the N–Z curve at the same viscosity indicates the speed required when operating at the maximum pressure. Suppose, for example, an oil having a viscosity of 5 centipoises at 250 deg. F. is used in this bearing. From Fig. 3 the values of P and N at $Z=5$ are 374 lb. per sq. in. and 2,260 r.p.m. respectively. Thus if the bearing were operating with this oil at a speed of 2,260 r.p.m. and a load such that the pressure on the projected area was 374 lb. per sq. in. it would come to equilibrium under the assumed conditions when the temperature reached 250 deg. F. and the value of ZN/P would equal 30. It is obvious that if N_1 and P_1 are a pair of limiting values of N and P, any

value of P lower than P_1 may be combined with N_1 without reducing the value of ZN/P below the permissible limit or increasing the value of ΔT above the permissible limit. Thus a bearing which can be operated at N_1P_1 can be operated safely at N_1 and any value of P below P_1. With a value of P lower than P_1 for a given oil the bearing will not be operating at ZN/P=30 when the limiting temperature is reached; thus the value for PN of 850,000 no longer holds and some value of N other than N_1 becomes the limiting value. This new limiting value may be obtained by the substitution in equation (7) of the limiting value of ΔT, the value of Z at the limiting temperature and the given value of P. Equation (7) will then take on the form $a\mathrm{N}[b\mathrm{N}+\Delta f]=\mathrm{K}''$, where a, b, Δf and K″ are constants, and thus may be solved directly for N. Conversely, if the pressure is held at P_1 and N is less than N_1 the value of PN will be lower than the limit. However, the question whether the value of ZN/P is within the limit, depends upon whether the lower operating temperature will cause an increase in Z great enough to counteract the decrease in N; and hence for a given bearing it will depend upon the viscosity-temperature relation of the particular oil used. The curves in Fig. 3 also provide a means for selecting a lubricant that will tend to provide maximum bearing capacity for the particular operating conditions. They indicate, for example, the desirability of the use of a low-viscosity oil where high speed is the limiting factor and a high-viscosity oil for high-load conditions.

The solid curve in Fig. 2 indicates the limiting values of PN at various values of ZN/P, all other factors remaining as given above. The magnitude of the effect at low values of ZN/P of the constant Δf is shown by comparing this curve with the dotted curve, which is based upon the Petroff equation where the journal and bearing are assumed to be co-axial. Equation (7) is based upon friction data obtained with bearings that were not run in and is not especially applicable for low values of ZN/P near the minimum point of the f–ZN/P curve. Data (McKee 1927–8, 1932) obtained on well run-in bearings indicate that, with some bearings, the friction at low ZN/P values may be lower than given by equation (7). With well run-in bearings, therefore, the limiting values of PN at low values of ZN/P may lie between the two curves in Fig. 2.

Too much weight should not be given to the actual values obtained in this numerical example. It is not based upon a bearing in actual operation, and the limiting values assumed are not necessarily the best for any particular application. Experimental data indicate that the minimum point of the f–ZN/P curve may occur at ZN/P values from 1 to 50, depending upon the kind of bearing metal, the machining of

the bore, and the amount of running-in. The value 30, therefore, would probably be applicable for a bearing that has had a reasonable amount of running-in. The limiting temperature 250 deg. F. has no particular significance; it possibly approaches the temperature range where the question of oil stability becomes significant and where the hardness and compressive strength of some of the tin-base bearing metals (Herschman and Basil 1932) may be reduced by as much as one-third to one-half. The heat dissipation coefficient chosen lies toward the upper range of some experimental data (Karelitz 1930, Hersey 1936) and represents conditions of high convection currents.

Conclusion. A method is suggested as a possible basis for design, more especially for heavy-duty, high-speed bearings. The method is directly applicable to self-cooled bearings, and may be made applicable to independently cooled bearings either by the insertion in equation (7) of suitable factors relating to the characteristics of the cooling system or in its present form to indicate a limit beyond which independent cooling is necessary and thus provide a measure of the capacity of the cooling system required. With the data now available it will provide only a rough approximation of bearing capacity. Further steps required before it could be applied more rigorously are a complete investigation of the frictional characteristics of journal bearings at the lower values of ZN/P and a comprehensive study of the factors affecting heat dissipation for various types of bearing applications.

NOTATION

D Journal diameter, inches.
L Bearing length, inches.
C Running clearance (difference between bearing diameter and journal diameter), inches.
W Total load acting on bearing, pounds.
N Speed of journal, revolutions per minute.
$V=\pi DN/12$ Peripheral speed of journal, feet per minute.
Z Absolute viscosity of lubricant at atmospheric pressure and bearing temperature, centipoises.
F Tangential frictional force, pounds.
$f=F/W$ Coefficient of friction.
$P=W/LD$ Pressure on projected area of bearing, pounds per square inch.
H Heat generated in bearing per unit time, British Thermal Units per minute.
H′ Heat dissipated by bearing per unit time, British Thermal Units per minute.

A Bearing area effective for heat dissipation, square inches.
T_B Bearing temperature, degrees Fahrenheit.
T_0 Temperature of surrounding atmosphere, degrees Fahrenheit.
$\Delta T = T_B - T_0$ Temperature rise.
$k_1 = \dfrac{1}{778 \times 12}$ Mechanical equivalent of heat, British Thermal Units per inch-pound.
Δf Correction factor, function of length-diameter ratio.
k_2 Coefficient in equation (3)$=473 \times 10^{-10}$.
k_3 Overall coefficient of heat transfer.
$k_4 = A/\pi DL$ Ratio of effective area for heat dissipation to the working area of the bearing.

References

Barnard, D. P. 1932 S.A.E. Journal, vol. 30, p. 192, "A Possible Criterion for Bearing Temperature Stresses".

Herschman, H. K., and Basil, J. L. 1932 Proc. Am. Soc. for Testing Materials, vol. 32, part 2, p. 536, "Mechanical Properties of White Metal Bearing Alloys at Different Temperatures".

Hersey, M. D. 1915 Trans. A.S.M.E., vol. 37, p. 167, "Laws of Lubrication of Journal Bearings".

Hersey, M. D. 1936 "Theory of Lubrication." John Wiley and Sons Inc., London and New York.

Karelitz, G. B. 1930 Trans. A.S.M.E., vol. 52, APM–52–5, p. 57, "Performance of Oil-ring Bearings".

McKee, S. A. 1927 *Mechanical Engineering*, vol. 49, p. 1335; 1928 *Mechanical Engineering*, vol. 50, p. 528. "Effect of Running-in on Journal Bearing Performance".

McKee, S. A., and McKee, T. R. 1929 Trans. A.S.M.E., vol. 51, APM–51–15, p. 161, "Friction of Journal Bearings as Influenced by Clearance and Length".

McKee, S. A., and McKee, T. R. 1932 S.A.E. Journal, vol. 31, p. 371T, "Journal Bearing Friction in the Region of Thin Film Lubrication".

SOME FACTORS THAT MAY DETERMINE THE SERVICE LIFE OF TIN-BASE BEARING METALS

By D. J. Macnaughtan*

Tin-base alloys have a long history of successful use as bearing metals for conditions of severe service in a wide variety of engines and machines. In recent years in certain types of internal combustion engines, where the loading on the bearings is very severe, trouble has arisen due to cracking and breaking up of the white bearing metal, after relatively limited periods of service. Increased knowledge of some of the conditions that cause the cracking has made it possible in a number of cases to reduce its occurrence by suitable modifications in engine design and operating conditions. A complete understanding of the nature of the failure is, however, still lacking. Fuller knowledge is desirable as this may suggest further possibilities of improvement to the metallurgist and engineer, whereby the excellent running properties of the tin-base alloys may be utilized in these particularly severe conditions of service.

The Nature of the Failure. Micro-examination shows that the cracks start at the surface of the whitemetal, work downwards well into the metal, and then frequently turn sidewards before reaching the bond. Willis (1937) suggests that the cracks follow the boundaries of the macro-grains. This may sometimes happen, but detailed examination of many examples of cracked bearings does not support the view that there is any close relationship between the cracks and the macrostructure. Fig. 1 shows the penetration of a crack from the surface into the body of the whitemetal. Fig. 2 shows a crack which has worked through the metal, turned sideways, and advanced parallel to the bond, although not along the bond itself. This appears to happen quite frequently, although failure along the bond may occur when the whitemetal has not been initially highly adherent to the shell. The combined effect produced by cracks which have proceeded downwards from the surface into the metal and then turned sideways, ultimately results in the detachment of pieces of the bearing metal. This causes failure owing to increase of friction and overheating as a result of the disruption of the surface and the consequent faulty flow of oil. These features of the defect have resulted in a general acceptance of the view that the failure by cracking is not primarily due to brittleness of the bond, but to fatigue cracking of the whitemetal itself.

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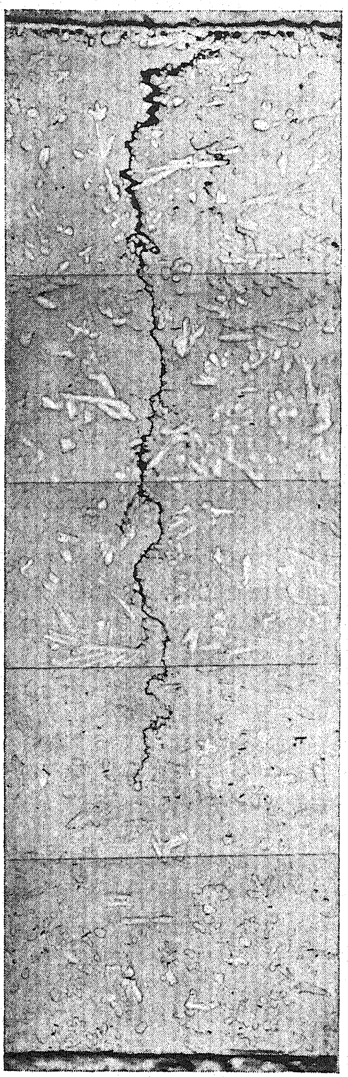
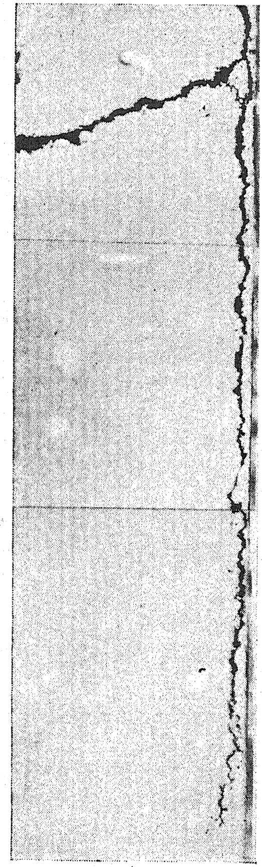
×250 dia.



Steel liner

Fig. 1. Cross-Section of a Cracked Whitemetal Coating, showing a Typical Radial Crack

×250 dia.



Steel liner

Whitemetal

Fig. 2. Cross-Section of a Cracked Whitemetal Coating

The vertical crack has forked and continued parallel to, but just above, the bond.

The Fatigue Strength of Tin-Base Bearing Metals. Results have been published which show the fatigue strength at room temperature of tin-base alloys with an average copper content of 3–4 per cent and various amounts of antimony up to about 12 per cent (Macnaughtan 1934; von Göler and Pfister 1936). Increase in fatigue resisting properties is obtained as the antimony content is raised to about 7 per cent, above which the increase is only slight.

There does not appear to be any published information on the fatigue properties of white bearing metals at the temperatures at which failure by cracking is most liable to occur in service, i.e. in the range 100–150 deg. C. As an alternative to the lengthy endurance tests that are required to provide this information, it appears possible that the change

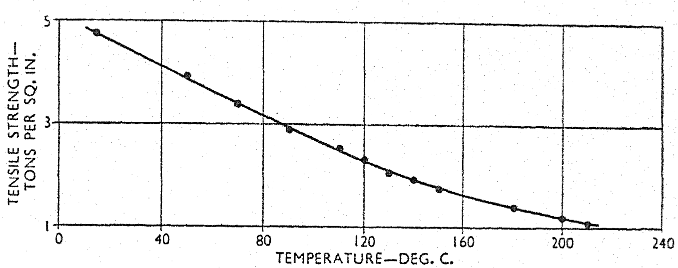


Fig. 3. Tensile Strength–Temperature Curve for Tin-Base Alloy containing 7 per cent Antimony, 3·5 per cent Copper

of tensile strength with temperature may prove a fair guide, as there is a general relationship between the tensile strength and fatigue properties at room temperatures. Information is now available on the change of tensile strength from 20 to 150 deg. C. for a series of tin-base bearing metals (Homer and Plummer 1937; Greenwood 1937).

A typical curve showing the change of tensile strength with temperature of a tin-base alloy containing 7 per cent of antimony and 3·5 per cent of copper, is reproduced in Fig. 3. The fatigue strength of this alloy, determined in the Haigh fatigue machine, was 2·07 tons at 20 deg. C., and 0·78 tons at 150 deg. C. on a basis of 20 million cycles. Thus, the alloy retained 37·7 per cent of its fatigue strength at the higher temperature and 38 per cent of its tensile strength. Results of fatigue tests, kindly supplied to the author by Mr. C. G. Williams, of a similar alloy by the rotating cantilever method show a fatigue strength at 130 deg. which was 46·8 per cent of that at 20 deg., whereas over the same range of temperature the tensile strength fell to 44 per cent of its original strength. These results, as far as they go, show, for a typical tin-base bearing metal, a fair relationship between the change of ten-

sile strength and fatigue strength with increasing temperature up to 150 deg.

Fatigue Stresses in a Bearing. When the conditions of loading in a bearing are contrasted with those produced in an ordinary fatigue test there are difficulties in making direct comparison. The nearest approach to similarity in respect of the nature of the stresses imposed on the whitemetal is given by those produced by flexing, as this causes the bearing shell to open and close in a cyclic manner, whereby alternating tension and compression are produced in the whitemetal. If every failure of whitemetal by cracking were due to flexing, the results of laboratory fatigue tests would have a direct relationship to the nature of the failure. Although flexing of the bearing shell is frequently a contributory cause, and in special cases may be the sole cause of the trouble, it is not the major cause of failure in many instances. Thus, Ricardo (1937) cites an example in which cracking of the whitemetal in the big-end bearings in a compression-ignition engine was eliminated by reducing the rigidity of the bearings, whereby flexing would, if present, be actually increased. The original failure occurred under conditions in which the whitemetal was subjected to pulsating pressure in a rigid support. These conditions impose a range of cyclic loading very different in character from that applied in the usual fatigue test. Thus, whereas in the test the metal is subjected not only to compressive stresses but also to tensile stresses, usually equal in amount, the normal action of the loading on a rigid bearing produces cyclic compressive stresses, but does not in any direct manner apply tensile stresses. In the absence of tensile stresses, it is difficult to see how cracks may be produced or at least opened up and enlarged so as to causê failure. However, although tensile stresses are not directly applied they nevertheless may be present. The following possibilities have been discussed by the author (1934):—

(1) Although only pressure is initially produced in the whitemetal in the lateral and circumferential direction by applied pressure through the oil to the surface of the metal, it would appear that as a result of the local spread of the metal around the area subjected to maximum pressure, circumferential and longitudinal tensions may be produced in the metal. These tensions, however, would diminish to zero on removal of the applied pressure if this does not exceed the elastic limit of the metal, although residual tensions would remain if the elastic limit had been exceeded. Actually, however, these tensions are produced around and outside the actual zone of maximum pressure in which there is the greatest tendency for cracking to occur.

(2) The friction of the oil upon the surface of the whitemetal,

although chiefly transmitted by shear stress in the whitemetal, might conceivably produce circumferential tension. The magnitude of the tensile stresses thus produced, however, cannot be large. Assuming a peak oil pressure as great as 6,000 lb. per sq. in. and a coefficient of friction equal to 0·015 (a high value), the tangential stress along the surface of the whitemetal is only of the order of 90 lb. per sq. in.

(3) Tension set up in the whitemetal as a result of the considerable difference between the coefficient of thermal expansion of the whitemetal and the steel shell upon which it is cast. A fall of temperature of 100 deg. would produce 6·8 tons tensile stress in the whitemetal if it remained elastic. Actually, of course, the metal yields at a lower stress. In reality, a fall of temperature a little greater than 50 deg. would produce tensile stresses which reach the yield point of any ordinary whitemetal.

It was concluded from the foregoing that the chief source of tensile stress in the whitemetal was due to possibility (3). Tests subsequently carried out for the author, which reveal the degree of stress in a whitemetal lining by the changes in dimension of a thin steel shell coated with the whitemetal, show that the stress caused by difference in thermal contraction may be relieved by creep of the whitemetal to a degree determined by the temperature involved. Thus, the stress in the whitemetal after casting on to the steel shell and cooling to room temperature is found to be high and of the order of magnitude of the yield point of the metal. Upon remaining at room temperature this stress is only slightly reduced by creep. When, however, the whitemetalled shell is subsequently heated to 150 deg., the stress in the whitemetal soon drops to a small value, owing to rapid relief of stress by a high rate of creep. The stress falls in this manner to a value which is probably near the limiting creep stress of the whitemetal at 150 deg. For a typical tin-base bearing metal (7 per cent antimony, 3·5 per cent copper), this is only of the order of 190 lb. per sq. in. On subsequently cooling down to room temperature the stress in the whitemetal regains its original high value. It would thus appear that during starting and initial running before a bearing has warmed up, the tensile stresses will be high, but they become small when the bearing is running warm. The tensile stresses due to difference in thermal contraction may thus have an important effect upon the life of a bearing subjected to very intermittent running. It is, however, well known that cracking will start and develop during continuous running when the temperature of the whitemetal keeps at a fairly constant high value. Under these conditions the tensile stresses in the bearing due to all the above-mentioned causes are very small in amount, compared with the

tensile stress, namely, 0·78 ton, that is required to cause fatigue in the laboratory tests at 150 deg. C.

There is a possibility, however, that these tensile stresses may be considerably augmented as a result of variations in temperature of the surface of the whitemetal when the bearing is running in a heated condition. Thus, Russell (1934) suggested that cycles of heating and cooling of a local area of the surface of a bearing metal would produce tensile stresses that would tend to cause and open up cracks in the area involved. Subsequently Russell* has shown that a network of cracks can, in fact, be produced when a portion of the surface of a whitemetal is subjected to a relatively small number of cycles (50 to 100) of rapid heating nearly to the melting point of the metal, followed by rapid cooling. This raises the question whether variations in temperature of this order may be produced during the running of a bearing in the portion of the surface where the variation in intensity of loading is most severe.

If at the moment of maximum pressure the oil film is partially penetrated, the surface temperature in the area involved might be expected to rise considerably, as indicated by the work of Bowden and Ridler (1936). Rapid cooling would follow by the inrush of relatively cool oil as the pressure falls. The loading required to produce these conditions would have to be extremely severe.

It is possible, however, that important effects due to temperature variations may be produced although fluid lubrication is continuously maintained. Bailey has stated (1934): "The oil film is continuously delivering energy, but at a varying rate, and therefore at the surface of the metal there may be variations of temperature much larger than might be imagined. An estimate of that effect, however, shows the variations of temperature of the metal to be less than 1 deg. C." This calculation, however, may not have taken into consideration the large variation in the intensity of loading that is imposed on bearings which give trouble by cracking in high-speed carburettor engines and in compression-ignition engines. Even, however, if there is only a small rise and fall of temperature the stresses produced in the surface of the whitemetal may be considerable. Thus, if a very thin layer of whitemetal suffers a change of temperature while the rest of the whitemetal and steel liner remain at constant temperature, a rise of 1 deg. C. will produce about 300 lb. per sq. in. of compression, and a superficial fall of temperature of the same amount produces about 300 lb. per sq. in. of tension. Thus, a range of variation of only 2 deg. C. produces an alternating stress corresponding to a range of stress of 600 lb. per sq. in. In view of the foregoing, it seems desirable to know what

* Written communication to the author.

variations in temperature do actually occur at the surface of the white metal under conditions of loading that lead to failure by cracking.

Development of Tin-Base Bearing Metals of Improved Fatigue Properties. The foregoing survey of the general nature of the stresses that may cause cracking of the whitemetal does not modify the view previously set forth (Macnaughtan 1934) that ordinary methods for fatigue testing, employing alternating stresses, are likely to yield valuable data for the development of improved alloys for use in severe conditions of service. An important question arises as to the temperature at which the fatigue properties should be compared. This may be illustrated by a reference to the influence of small additions of cadmium to improve the fatigue properties of tin-base bearing metals and the effects of small amounts of lead which are liable to be present as an impurity in the alloys.

(1) *The Effect of Cadmium.* Temperature–tensile strength curves show that at all temperatures up to 150 deg. there is an increase in strength as the amount of cadmium added to a tin-base alloy containing 7 per cent of antimony and 3·5 per cent of copper is increased up to about 3 per cent (Homer and Plummer 1937). Consistent with a general relationship between tensile strength and fatigue strength, the results of fatigue tests carried out for the author by Professor Haigh show an increase in fatigue strength at 150 deg. of 13 per cent with 1 per cent cadmium, and 21 per cent with 2·5 per cent cadmium, for an endurance of 10 million cycles to fracture. If, however, a higher temperature is involved the order of merit may be altered. Thus, temperature–tensile strength tests show a falling off in strength of the 3 per cent cadmium alloys relative to those of lower cadmium content when the temperature rises above 150 deg. and the strength of this alloy becomes actually less than that of the alloy containing 1 per cent cadmium when the temperature reaches 180 deg. C. This is probably due to the presence of a eutectic of low melting point.

(2) *The Effect of Lead.* In Fig. 4 are shown the tensile strength–temperature curves of alloys of 7 per cent antimony and 3·5 per cent copper with and without 1 per cent cadmium, and with the addition to each alloy of 0·5 per cent lead. At 150 deg. C. the alloy containing cadmium has a tensile strength about 20 per cent greater than the alloy without cadmium. At temperatures above 150 deg. the presence of a low melting point complex eutectic containing lead causes a rapid drop in the strength of the alloy containing cadmium, with the result that above 170 deg. C. its strength is actually lower than that of the other alloy.

In the absence of precise knowledge of the actual temperature

reached at the surface of whitemetals under conditions of severe loading, the safest deduction would be to assume that the permissible limit for the cadmium addition in this tin-base alloy is about 2 per cent and that the alloy should be as far as possible free from lead.

The author desires to thank Professor B. P. Haigh, M.B.E., D.Sc., M.I.Mech.E., for valuable discussions on matters dealt with in this

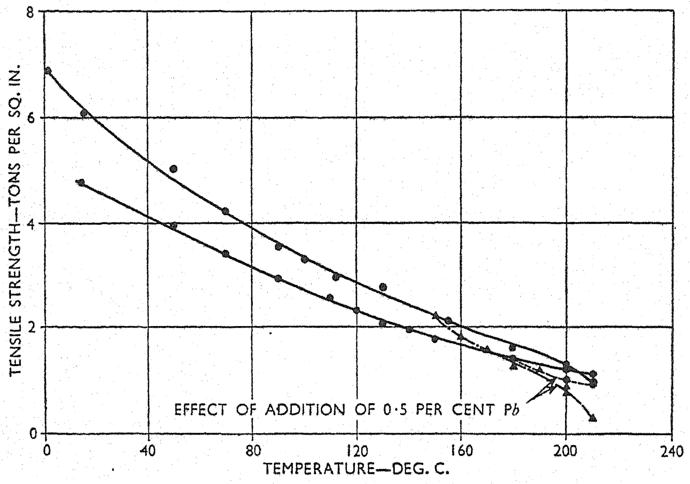


Fig. 4. Tensile Strength–Temperature Curves for Tin-Base Alloys of the Percentage Composition Given Below

Upper curve: Sb, 7; Cu, 3·5; Cd, 1.
Lower curve: Sb, 7; Cu, 3·5.

paper. The experimental work referred to, in which the author has had the assistance of his colleagues, is part of a programme of research for the International Tin Research and Development Council, to whom thanks are due for permission to publish.

References

Bailey, R. W. 1934 Discussion on Macnaughtan's 1934 paper (see below).

Bowden, F. P., and Ridler, K. E. W. 1936 Proc. Roy. Soc. A, vol. 154, No. 883, p. 640, "Surface Temperature of Sliding Metals".

GREENWOOD, H. 1937 Technical Publications, International Tin Research and Development Council, A, No. 58, "The Tensile Properties of a Series of Whitemetal Bearing Alloys at Elevated Temperatures".

HOMER, C. E., and PLUMMER, H. 1937 Technical Publications, International Tin Research and Development Council, A, No. 57, "Mechanical Properties of some White Bearing Metals and other Tin-Base Alloys at Various Temperatures".

MACNAUGHTAN, D. J. 1934 Jl. Inst. Metals, vol. 55, p. 33. "The Improvement of White Bearing Metals for Severe Service".

RICARDO, H. R., and PITCHFORD, J. H. 1937 Soc. Automotive Eng., May, preprint, "Recent Trends and Developments in European Automotive Diesel Engine Design".

RUSSELL, H. W. 1934 Discussion on Macnaughtan's 1934 paper.

VON GÖLER, K., and PFISTER, H. 1936 *Metallwirtschaft*, vol. 15, pp. 342, 365, "High-Tin White Bearing Metals".

WILLIS, A. B. 1937 *Machinist*, vol. 81, p. 268E, "Developments in Bearings and Bearing Materials".

TILTING-PAD BEARINGS AND THEIR PRACTICAL LIMITATIONS

By A. G. M. Michell, F.R.S.*

The purpose of the present paper is to review briefly the development of bearings of the tilting-pad type and to point out the possibilities of their future improvement and extended application, having regard to their constructional limitations.

The initial objective in the origination of the tilting-pad type of bearing was the production of a thrust bearing which should be film-forming in the same way as the Tower-Reynolds journal bearing, and be equal to it in load-carrying capacity and efficiency. That objective was immediately reached by the application of the tilting pad in accordance with the theory which the author derived from Reynolds's principles and first published in 1905.

In the applications which were made of the principle in its first few years, which were mainly to centrifugal pumps and water turbines, the intensities of bearing pressure and shaft speeds were similar to those at that time usual in the journal bearings of such machines. A mean pad pressure of 50 atmos. and a peripheral speed of 5 metres per sec. were rarely much exceeded. Practical experience of these bearings amply demonstrated the theoretical characteristics in which they differed from the thrust bearings previously in use, namely, that increase of intensity of pressure (other conditions being unaltered) raises the efficiency of the bearing, and that higher speed increases its safety of operation by increasing the thickness of the oil film.

That the possibility and advantages of the tilting-pad principle are not limited to thrust bearings but extend also to journal bearings was apparent to the author from the beginning, and successful applications of this class were made at an early stage. Through accidental circumstances, however, the marine thrust bearings became more widely known than any other applications of the principle, and it came to be generally inferred from the comparatively low specific loading usual in this application, that little or nothing was to be gained by replacing Tower-Reynolds journal bearings by bearings of the pivoted type. Yet it is undeniable that from about 1915 onwards the development of highly stressed machinery, including steam turbines, large electric generators, and internal combustion engines, especially those for flying machines, and also Diesel engines, has been hampered by the limitations of the journal bearings in use. This has been particularly the

* Melbourne, Australia.

case in multicylinder internal combustion engines, where economical spacing of the engine cylinders necessarily limits the axial length available for the shaft bearings. Such bearings have in consequence to be loaded so highly as to compromise their factors of safety against unduly rapid wear and shaft fractures arising from that wear.

In large turbines and similar machines there is not the same pressure upon the manufacturer to reduce the axial dimensions of the bearings, since these bearings are located at the ends and not in the interior of the machines. Moreover, the journal speeds are higher than in reciprocating engines and the lubricating films thicker in consequence, so that wear is less rapid, if not eliminated altogether. Accordingly, Tower-Reynolds journal bearings continue to be used in these machines, and by restriction to low intensities of pressure the frequency of their necessary refittings is kept within tolerable limits. Where they have been replaced by tilting pads, however, the result has been increased efficiency and reduction of the axial lengths of the bearings to about one-third, with a large consequent saving in the overall dimensions of the machine and its house room.

It is well known that, as regards film lubrication under heavy loading, there is in the Tower-Reynolds journal bearing the inherent defect that a converging and pressure-generating film can only be formed on a small portion of the circumference of the bearing, namely, about 45 deg. of arc, which, in usual practice, is specially fitted to the journal by hand. The permissible intensity of load on the journal, calculated on its full diametral area, is consequently only about one-third of the mean intensity of load on the effective bearing area. In other wôrds, it is necessary, in order to carry a load equivalent to a diametral mean pressure of 150 atmos., as present practice in internal-combustion engines often requires, to subject the effective bearing surface to a mean pressure of about 400 atmos. The maximum intensity on this area consequently approaches very near to the crushing strength of the bearing metals commonly used, and the accuracy of fitting necessary to prevent local metallic contact is apt to be higher than manufacturers can provide under commercial conditions.

Another disadvantage arising from the same characteristic of the cylindrical bearing is that it is only capable of producing an effective tapered film with one particular direction of application of the load, for which it must be fitted in advance, whereas in almost all applications the direction of the load on a journal bearing is continually varying, and, even if constant, is seldom assignable with much accuracy. Thus in reciprocating engines, not only is the direction of the loading on the crankshaft bearings constantly varying throughout the stroke,

but the pattern of this variation varies with every change of speed. In the bearings of marine shafting, such variations may occur through the straining of the ship in a seaway, whereby unknown loads may be imposed on any part of the bearing or its cap.

Having such defects, it would seem that the cylindrical journal bearing must ultimately be restricted to applications in which its simplicity and, in small sizes, its comparative cheapness outweigh all deficiencies.

In its earliest stages the tilting pad for journal bearings was regarded mainly from the point of view of its being a single-pad bearing and thus merely a self-adjusting substitute for the hand-fitted load-carrying segment of the Tower-Reynolds bearing. It soon became apparent, however, that further advantages would be obtained if tilting segments were arranged in a series all round the journal. It is this symmetrical

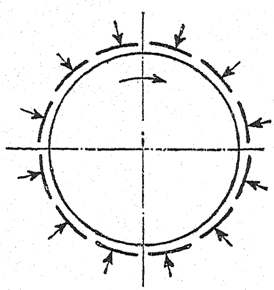


Fig. 1. Symmetrical Multi-pad Journal Bearing

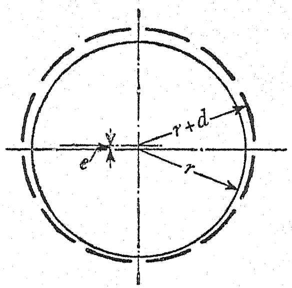


Fig. 2. Loaded Multi-pad Journal Bearing

multipad form of journal bearing, shown in outline in Fig. 1, which is referred to as the "tilting pad type" in the remainder of the present paper. Its theoretical characteristics are analogous to those of a tilting-pad thrust bearing whose collar runs with very small clearances between two opposed series of pads, so that fluid pressures are generated on both sides of the collar and the resultant thrust is changed from one axial direction to the other by a very small change in the axial position of the collar with respect to the pads.

From practical points of view, on the other hand, the tilting pad journal bearing presents analogies with ball and roller bearings, particularly in its adaptability to accurate machine manufacture without the particular purpose or destination of any bearing being known in advance. In order that they may serve all probable requirements, besides being adapted to replace cylindrical bearings in cases where the latter have failed, such "general purpose" bearings must be capable

with suitable lubricants, of carrying loads of 100 atmos. or more on the projected area of the journal collar, with peripheral speeds of 20 metres per sec. and upwards, and there should be no limit on the lengths of their working lives.

When the pads of such a journal bearing are, as is desirable, at least eight in number, each pad extending through not more than 45 deg. of the circumference, near approximations to its operating characteristics can be expressed in quite simple formulæ derived by treating the number of pads as indefinitely large. As the journal revolves within the pads of such a bearing, supplied of course with a sufficiency of oil, fluid pressures are exerted between each pad and the opposed surface of the journal. If there is no load on the journal these pressures are equal on all the pads, and the journal is in equilibrium when running concentrically within them, as shown in Fig. 1. When the journal carries load, the thickness of the oil film becomes slightly reduced on the pads towards which the load is directed, and correspondingly increased on the opposite side of the bearing, as indicated in Fig. 2. The differences of film thickness, which involve only a very small displacement of the journal from its central position, increase with increasing load but diminish with increasing speed. The load is, of course, equilibrated by the fluid pressures, which become greater on the side of the bearing towards which the journal is displaced than on the opposite side. There is a very simple relation between the amount of the journal load and the sum of all the radial pressures, namely, that they are to one another in the ratio of the displacement e of the journal from the central position to the radial clearance d between the journal and the ring of pads when the journal is central.

The journal load per unit of axial length is given in terms of the viscosity of the lubricant and the dimensions and speed of the journal by the formula

$$\overline{\mathrm{B}}=\mathrm{C}\times\mu\mathrm{U}r^2e/n(d^2-e^2)^{\frac{3}{2}}$$

in which μ is the viscosity, U the peripheral speed, and r the radius of the journal, n the number of the pads, and C a constant depending on the shape of the pad and the location of its pivoting point. This formula, now published for the first time, as far as the author is aware, is fairly approximate when n is at least 8 (as it should be), and becomes more and more exact for larger values. The quantities involved must all, of course, be expressed in one rational system of units. In the C.G.S. system the constant C has the value 14·1 if the pads are pivoted behind their centres in the normal manner and their axial dimensions are large compared with their circumferential lengths. The corresponding constant is 5·9 if the effective surfaces of the pads are square.

For other proportions the value of C may be taken from the literature of the plane pad.*

The same remarks apply to the formula for the total frictional resistance exerted on the journal, namely

$$\overline{F}=6{\cdot}535\cdot\cdot\times\mu Ur/(d^2-e^2)^{\frac{1}{2}}$$

and to the coefficient of friction of the bearing, which is

$$f=\overline{F}/\overline{B}=0{\cdot}463\cdot\cdot\times(n/re)(d^2-e^2)$$

the numerical constants corresponding to C=14·1.

All these formulæ will, of course, equally apply if the ring of pads rotates about a stationary journal, or together with the journal within an outer stationary race as in Briggs's crankpin bearing (British Patent No. 211,707 of 1923). In the latter cases, however, due allowance must be made, in calculating the useful load, for the centrifugal force corresponding to the centrifugal mass-accelerations of the pads.

It will be immediately seen from the formula for the journal load $\overline{B}$ that it increases very rapidly with the displacement of the journal from the central position, and that it approaches infinity as its limit when e becomes equal to d, or, in other words, when the clearance becomes very small on the loaded side. As this limit is approached, the useful load becomes more and more nearly equal to the sum of all the radial pressures on the pads, and at the same time the coefficient of friction theoretically diminishes to zero.

The shapes of the curves of the load, frictional resistance, and coefficient of friction will be seen from Fig. 3. In practice the ratio of e to d will usually be somewhat greater than $\frac{1}{2}$ under the heaviest loads for which the bearing is intended; and d, for bearings of moderate size, will be of the order of 0·001 cm., diminishing, however, for smaller bearings and for larger values of n. It will be readily seen that, with these values, heavy loads can be carried in any direction with displacements of the journal from the central position so small as to be negligible in practice in comparison with the elastic deformations of ordinary shafts and bearing pedestals.

The practical limitations of multipad bearings, both of thrust and journal types, assumed of course to be made with the same care as to materials and accuracy of workmanship as other bearings produced in quantity to standard dimensions, arise only from:—

(1) Insufficient strength or rigidity in the parts to maintain the theoretical forms with the necessary accuracy; or,

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* E.g. Kobayashi, "A development of Michell's theory of lubrication".

(2) Insufficient means for removing the heat (produced by the shearing of the oil film) and thus maintaining sufficient viscosity in the oil.

Since the pads around an entire semicircumference share the useful load, all but the most extreme requirements can be met without exceeding a mean intensity of pressure on them of 100 atmos., corresponding to $100 \times \pi/2$ or 157 atmos. on the projected diametral area, and, with $e=d/2$, to a mean pressure on the most heavily loaded pad of about 260 atmos. With such pressures special care must be taken to provide sufficient strength and rigidity in the portions of the pads

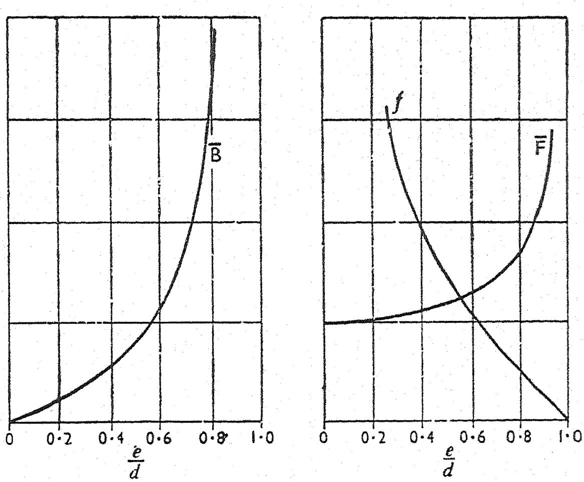

Fig. 3. Curves of Load B, Coefficient of Friction of the Bearing f, and Total Frictional Resistance $\overline{F}$

constituting the tilting surfaces, or pivots, and in the parts which support these pivots. In order that the pad may tilt about the predetermined axis, or one sufficiently close thereto, the pivot has to be limited to rather restricted dimensions. It can, in the forms of pads hitherto employed in tilting bearings, hardly be allowed to exceed about 10 per cent of the surface of the pad, having, say, one-fifth of its circumferential length and one-half of its axial length. The maximum intensity of pressure on this area, assuming flat contact surfaces, may be double the mean, and therefore 20 times as great as the mean fluid pressure on the pad, or 5,200 atmos. under the load assumed. Such a pressure is beyond the crushing strength of the materials ordinarily used. Even if hardened steel is used, the form

of the pivot usually precludes film lubrication, so that contact corrosion and slow abrasion must be expected to occur with such loads.

In the piston slippers of the author's crankless engines, which operate in respect to lubrication in precisely the same way as pivoted bearing pads, and which commonly sustain mean intensities of pressure as great as, or greater than, the 100 atmos. which have just been instanced, the construction of the pivot is such that it is film-lubricated and may have a projected area as great as one-half of the area of the pad itself.

The pivots of these slippers (Fig. 4) are hemispherical, this form being kinematically necessary to permit the slipper to follow the varying direction of the inclination of the slant, which corresponds to the collar in a thrust bearing, as well as being very desirable from the point of view of pivot lubrication. Such pivots run successfully for indefinite periods with loads up to 300 atmos.

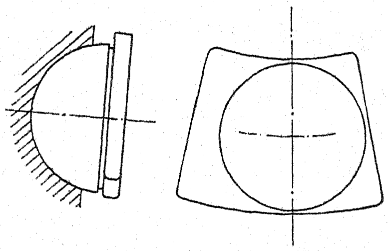

Fig. 4. Pivots of Piston Slippers

Failing the development of some comparable form of pivot for bearing pads, or some means of eliminating the pivot altogether, the maximum load capacity and life of tilting pad bearings must continue to be limited by the crushing strength of their pivots. The limitations of pivoted bearings as regards speed (or speed and load conjointly) depend, as in other bearings running at high speeds, on the means of removing the heat generated by the shearing of the lubricating films, in other words, preventing the viscosity of the oil from being so much reduced by rise of temperature, that metallic contact takes place. In this connexion it is to be remembered that in bearings for such loads and speeds as are now being considered, practically all the heat generated must be conducted away through the thickness of the film and through the metal parts of the bearing.

Thus with a bearing pad 10 cm. square working with a mean intensity of load of 100 atmos. and a sliding speed of 100 cm. per sec., the coefficient of friction being 0·001, the heat carried away by

the oil is only about 2 watts per deg. C. of the rise of temperature, while the heat conducted through the oil film in series with 2 cm. thickness of cast iron would be of the order of 20 watts per deg. C., the total amount of heat to be dissipated being about 980 watts. This may be dealt with, according to circumstances, by providing fins or a water jacket, or (as in the slant-slipper bearings of crankless engines) by spraying the moving parts with cool oil. It may be remarked that, according to the elementary theory of the lubricating film, the heat generated at a given speed varies directly as the square root of the intensity of the pressure, but the thickness of the film varies in inverse proportion to the same root. In so far as the dissipation of heat depends on conduction through the film, therefore, increase of load, *per se*, does not involve any rise of temperature. The limitation of the capacity of the bearing with respect to disposal of heat is therefore mainly with regard to speed, and from this point of view there is some advantage in so constructing the bearing that there are two lubricating films between the journal and the outer stationary member, as proposed by the author (British Patent No. 875 of 1905) but not hitherto adopted in actual practice. According to this arrangement, a ring of pads is so mounted as to be free to revolve about the axis of the bearing at approximately one-half of the speed of the journal.

THE LUBRICATION OF BEARINGS OF INTERNAL COMBUSTION ENGINES

By V. Mickelsen *

Oil Film Thicknesses. Theoretical and experimental studies of film lubrication now enable the coefficient of friction, eccentricity ratio, and minimum film thickness to be calculated within a tolerable margin of error, for bearings in which the load is constant in magnitude and direction. These two conditions are not fulfilled in most internal-combustion engine bearings and it is recognized that variation in the direction of loading is beneficial in preserving the integrity of the oil film.

It is well known that wear of the crankpins of four-stroke internal combustion engines is greatest on the side of the crankpin nearest the centre line of the shaft. This effect is particularly easy to observe on high-speed engines. A similar effect is noticeable on the main bearings of both four-cycle and two-cycle engines when the bearing lies between two unbalanced cranks on the same centre or with an angle of 90 deg. or less between them. In these instances the maximum wear is in line with the resultant centrifugal force which lies in the same direction as that of the maximum inertia in the neighbourhood of both top and bottom dead centres.

In such cases three conclusions seem legitimate:—

(1) The more intense but briefly sustained pressures due to combustion are of small importance compared with the less intense but longer sustained pressures due to centrifugal force and inertia.

(2) The wear (such as it is) is incurred during full-speed running and not when starting or stopping.

(3) Since continuous running under conditions of boundary friction is out of the question, the wear is presumably due to abrasive particles passing through the oil film at its thinnest part.

Whatever means are provided for the filtration or centrifugal separation of lubricating oil should be capable of eliminating solid particles smaller in diameter than the minimum film thickness of the bearings.

Conclusions (1) and (2) suggest that for purposes of calculation the resultant pressure on bearings most subject to wear may be represented

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* Chief Designer, Harland and Wolff, Ltd.

by a steady load equal to the centrifugal force of the revolving weight plus half the reciprocating weight of one set of parts multiplied by a factor taking into consideration the angle of spacing of the cranks each side of the bearing. This factor would be 1 for two cranks on the same centre and $\frac{1}{2}\sqrt{2}$ for two cranks at right-angles, and so on. On this basis the film thickness has been worked out for a number of typical instances shown in Table 1.

In the first six examples the crankshaft diameters vary from 11 to 29 cm., the speed of revolution from 1,200 to 300 per minute, and the equivalent centrifugal bearing pressures from 61 to 25 kg. per sq. cm. The minimum film thicknesses all lie between the limits of 0·5/1,000 inch and 1·25/1,000 inch. In the last two examples of crankshafts 34 and 43·5 cm. in diameter respectively, running at 170 and 120 r.p.m., on account of the use of balance weights the centrifugal pressures work out to low figures, namely, 15·5 and 12·7 kg. per sq. cm. and the corresponding film thicknesses are 1·3/1,000 inch and 1·5/1,000 inch. In the last two examples the loading due to gas pressure is probably more important than that due to centrifugal force and inertia.

Cylinder Lubrication. For trunk engine types it is the usual practice to use only straight mineral oil. Although this may not be exactly the best lubricant for cylinders in this type of engine, it is found preferable to use straight mineral oil, to avoid complications due to contamination. For engines in which the cylinders are separate from the crankcase, and in which it is now customary to use a straight mineral oil, it will no doubt be found that a compound oil is much more suitable, because in cylinders especially boundary lubrication is probable. Recent investigations show that a straight mineral oil, compared with a compound oil, is very deficient under these conditions, and no matter how finely the metal surfaces are finished it seems to be impossible with high pressures and temperatures to avoid boundary lubrication near the combustion space, resulting in the wear and tear which takes place with the materials used for liners and piston rings.

Bearing Lubrication. For bearings it is usual only to use a straight mineral oil, supplied under pressure through filters. Filters, however, can only collect the sludge and abraded particles as they are carried along with the oil stream, after they have wrought their harmful influence in the engine bearings. So it does not seem possible to diminish to any great extent the usual wear and tear of bearings by any method of filtration.

For an enclosed crosshead engine it may be possible, in the future,

TABLE. 1. FILM LUBRICATION OF TWO-CYCLE TRUNK ENGINE MAIN BEARINGS

Example	1	2	3	4	5	6	7	8
Diameter of crankshaft, cm.	11·0	12·5	15·0	18·0	22·0	29·0	34·0	43·5
Length of bearing, cm.	6·2	6·6	8·0	9·6	18·0	21·2	23·8	30·0
Speed, r.p.m.	1,200	1,000	800	600	375	300	170	120
Equivalent centrifugal bearing pressure, kg. per sq. cm.	53·0	56·5	61·5	38·3	25·9	24·6	15·5*	12·7*
Calculated coefficient of friction	0·0067	0·0058	0·0046	0·0053	0·0045	0·0041	0·0038	0·0034
Calculated minimum film thickness, thousandths of an inch	0·55	0·53	0·51	0·72	1·07	1·2	1·3	1·5

*Balance weights fitted.

to lubricate all the bearings with water or a water-oil mixture when all the bearings are filled with a composite material suitable for this type of lubrication. After all, in ordinary pressure oil lubrication the oil is used more for cooling than for ideal lubrication, and therefore gives rise to more waste.

Up to the present, experiments have only been carried out with lower surface speeds and pressures than those which are commonly used in actual practice, so it would be of considerable help to carry out further research and experiments on points such as the following :—

(1) Constant speeds with variable pressures for the same number of revolutions compared with constant speed and constant pressure.

(2) Influence of oil grooves at variable speeds and pressures.

COPPER ALLOYS AS BEARING MATERIALS

By D. P. C. Neave, M.A., A.M.I.Mech.E., and W. B. Sallitt, B.A., A.I.Mech.E.*

The traditional use of copper-rich bronzes as bearing materials for comparatively heavy pressures and slow speeds has been based so much on practical experience, never properly collected, that even the recent period of co-operative effort by engineers, metallurgists, and oil technologists, has scarcely served to codify quantitative information about this complex subject. Moreover, the recent exploitation of very high additions of lead has extended the use of copper-base bearings to the high-speed field, so that it is difficult in a short paper to avoid over-simplification.

Although an electrodeposited film of copper is used to enhance the running-in properties of hardened steel-to-steel parts subjected to severe pressures, such as automobile steering gears, nevertheless a solid copper bearing tends to "pick up" when running in contact with steel, and in general it is necessary to alloy copper to secure satisfactory bearing properties and adequate structural strength.

Copper-Tin Bronzes. Tin, usually in percentages from 5 to 15, has been pre-eminently successful as an alloying element for copper in bearing bronzes and, in comparison with most of the other appropriate materials, the tin bronzes show fair retention of strength and a low rate of oxidation at high temperatures, with good thermal conductivity. The recent report (Bowden and Ridler 1936†) that temperatures of the order of 600 deg. C. and 1,000 deg. C. may be attained at the surface of metals in sliding contact with boundary lubrication and dry conditions respectively, perhaps indicates that the success of the bronzes under severe bearing conditions may also be connected with certain little-known characteristics of their surface *vis-à-vis* steel at high temperatures.

α-Tin Bronzes. If the tin content is less than will give a 93/7 copper-tin ratio approximately, the exact ratio depending on the rate of cooling of the casting, tin enters into a simple (α) solid solution with copper. Bronze castings of somewhat higher tin content, such as 10 per cent, may also be homogenized into a simple α structure by annealing at 600 deg. C. or over. The α-tin bronzes work-harden fairly

* Copper Development Association.

† Proc. Roy. Soc. Series A, 1936, vol. 154, p. 640.

rapidly, and may be cold worked into tube or strip, for cutting off or rolling up into bearings. These cold-wrought bronzes, while convenient for thin-walled bushes, are not generally considered as good from a bearing point of view as the duplex cast tin bronzes; and it is conceivable that the finely distributed porosity which is characteristic of the cast tin bronzes may have some beneficial effect on lubrication. (The leaded α-tin bronzes are dealt with later.)

Tin Bronzes Containing Chromium, etc. Additions of the order of 2 per cent of chromium, usually with iron, have recently been made to the α-tin bronzes, giving a series of alloys which are reported to offer superior bearing properties under heavy-pressure and high-temperature conditions. These alloys can be hot-worked, and, as forged, they show the presence of hard iron-chromium compound in an α-copper-tin matrix.

α-δ Duplex Bronzes. If the tin content is greater than will give a 93/7 copper-tin ratio approximately, increasing amounts of a hard and brittle (δ) copper-tin constituent are present in the softer (α) matrix. Tin bronzes of duplex structure can neither be hot-worked nor cold-worked readily, so that fabrication is effected by machining from chill-cast bronze sticks, or, in the bigger sizes, from cored rods, from centrifugal castings, or from individually cast bushes or half-shells. The conventional duplex bronzes usually have a 90/10 or 89/11 copper-tin ratio. For very severe loading, such as for slide valves and turntable bearings, and for abrasive conditions, the wearing properties may be improved by increasing the proportion of the hard and brittle delta constituent, i.e. by raising the tin content, say, to an 85/15 or 80/20 copper-tin ratio, but the bronze then becomes brittle and difficult to machine.

Phosphor-Bronzes: Deoxidation. The tin-bronzes should be thoroughly deoxidized to avoid the presence of abrasive tin oxide particles likely to score the shaft. Deoxidation is usually effected by phosphorus, and the readiness with which tin oxide reacts with phosphorus or other suitable deoxidizers and slags off, may be one reason for the success of the tin bearing bronzes. If properly deoxidized, these bronzes will contain perhaps 0·05 per cent of residual phosphorus. Further addition of phosphorus (e.g. 0·5 per cent excess in "2 B8" phosphor-bronze) results in the formation of a hard copper-phosphide constituent. One per cent and more of phosphorus is sometimes added to heavily loaded bronze bushes and gearwheels, as an alternative to a higher, and more expensive, tin content; but some

authorities consider that the hard constituents arising from high phosphorus content cause undue shaft wear; moreover, the freezing range of the metal is increased considerably by phosphorus additions so that lead additions, if any, are more liable to segregate. (See under leaded copper-tin bronzes below.)

Gunmetals or tin-bronzes containing about 2 to 5 per cent zinc, have good casting qualities for housings and similar parts, and are sometimes used as bearing metals, but not extensively so in Great Britain, since their wearing properties are generally considered inferior to those of the phosphor-bronzes. As zinc is a deoxidizer, these alloys do not usually contain phosphorus.

Scope of the Rigid Bronzes. Since in most cases a viscous oil film cannot be maintained constantly, wear cannot be entirely eliminated, but it can be minimized by ensuring that the hardness of bearing and shaft are correctly related and that the bearing material has adequate plasticity to accommodate such imperfections of alignment, finish, and clearance as may exist. With hardened steel shafts, if the bearing is accurately aligned and well finished, and good lubrication is maintained, then the rigid bronzes hitherto mentioned may be used with advantage. These bronzes take an excellent surface finish, giving a low coefficient of friction and may permit some reduction in designed bearing area, while maintaining location of the shaft within very close limits. As shaft hardness, alignment, lubrication, or other conditions become less favourable, and as speeds mount, so it becomes increasingly desirable to use a leaded bronze.

Leaded Copper-Tin Bronzes. The other alloying elements mentioned here form solid solutions with copper, but an element of outstanding usefulness in the bronzes is lead, which is virtually insoluble in copper and its alloys. Lead additions of 5 to 25 per cent are made primarily to act as an emergency lubrication expedient when oil feed is uncertain or local pressures high, because the lead phase, which melts at about 314 deg. C., will sweat out on the bearing surface at high working temperatures, so minimizing the tendency to seizure and scoring of the shaft. To aid machining, about 2 per cent of lead is also added to bronzes and gunmetals when some loss of toughness is of no moment, as the lead particles cause the chips to break off short on the tool.

It is not easy to ensure the uniform distribution of lead as fine particles and to avoid segregation, and it is often desirable to cause the copper-tin constituents to freeze as quickly as the given mass of casting will endure without cracking. About 2 per cent of nickel is an optional

addition, especially for large castings, to aid in securing even distribution of lead.

The α-δ-bronzes do not usually contain very high lead additions, and a well-known composition of this type is 80 per cent copper, 10 per cent tin, and 10 per cent lead. Owing to its high tin-copper ratio, this alloy offers a resistance to compressive loads and to repeated pounding tests which is comparable with that of an "88/10/2" gunmetal.

Leaded α-Tin Bronzes; Plastic Bronzes. If the tin content is lowered to the α range, the strength and resistance to pounding of the bronze are reduced and its plasticity (unless work-hardened) is increased, whilst lead additions further reduce average structural strength. The α-bronzes have a shorter freezing-range than the duplex bronzes, so that lead, being less apt to segregate, can be added in amounts up to 25 per cent or even higher, giving the so-called plastic bronzes, which are employed where it is desirable to accommodate more readily small errors in fit and alignment, and so avoid local high pressures. An alloy of this type contains 70 per cent copper, 5 per cent tin, and 25 per cent lead. The wrought α-tin bronzes seldom contain high lead additions, as reheating or annealing cause trouble due to segregation of lead.

Bronzes for High Rubbing Speeds. As the tin content is lowered or the lead content raised, there is less tendency to score the shaft, and for high running speeds, it is generally desirable to use as high a lead content as conditions of loading will permit. This lead content can, in general, only be determined by experience, since alternating loads and couples may induce local stresses far in excess of the calculated average. With high rubbing speeds, particularly when the bearing is not exposed to high temperatures or severe abrasion, it has been a custom to employ bronze or gunmetal shells merely as a second line of defence, and to line them with tin-base whitemetals, especially in the larger diameters, as for dynamo rotor bearings, locomotive axle bearings, etc. Similarly, in the face of still more exacting combinations of speed and load, the plastic bronzes give way to steel-backed copper-lead bearings.

Copper-Lead Bearings. The "75/25" copper-lead bearing (D.T.D. 229) may be described as a high-leaded α-tin bronze in which the tin content has been reduced to about 1 per cent or less. To reduce wear on unhardened shafts, it is generally considered desirable to omit the tin content altogether, and the lead content may then be as high as 35–45 per cent with advantage. The lead additions apparently break up or smear over the copper matrix to give a satis-

factory surface polish. Where very heavy duty is imposed and close maintenance of bearing clearance is needed, as in automobile crankshaft bearings, the thickness of the copper-lead lining after machining must be reduced, perhaps to 0·020 inch, to avoid risk of spreading. In practice, the lining is metalled on to a mild steel shell in a non-oxidizing atmosphere at temperatures of the order of 1,200 deg. C., which would tend to melt a bronze liner and draw the temper of heat-treated steel. To aid in lead diffusion, up to 2 per cent of nickel (like 1 per cent of tin) is sometimes added to copper-lead bearings, especially of large mass, but such additions are very prejudicial to the otherwise excellent thermal conductivity of the copper matrix, and in general, it appears better to keep additional elements as low as possible.

To obtain the strong bond between copper-lead and its steel shell, gas absorption (likely to cause porosity of the bond) should be avoided, and attention must be paid to cleanliness of the shell and other points. The cost of manufacturing copper-lead bearings is, therefore, at present considerably higher than that of tin-base bearing metals (or Babbitts). Copper-lead can, however, withstand about 25 per cent higher loading, and higher working temperatures, but being less plastic, requires either about 50 per cent greater initial clearance than tin-base metal, or higher oil pressures. At oil temperatures above the boiling point of water, some differential washing-out of lead from the copper matrix has been reported with oils to which fatty acid has been added to enhance adsorption. However, the greater attention now paid to oil cooling in most high-duty applications can, it is believed, normally be relied upon to prevent this or similar effects from acid formation due to oxidation.

Porous Powder-Moulded Copper-Tin Bearings. For medium-duty application, chiefly in sizes up to about 3 inches in diameter, fabrication can be simplified by moulding bronze bushes direct from a mixture of about 90 per cent copper powder and 10 per cent tin powder, sometimes with graphite added, and sintering at temperatures of the order of 700 deg. C. in a non-oxidizing atmosphere. These bearings have a structure resembling that of the cast duplex tin-bronzes, but do not at present contain lead. By adjusting moulding conditions, the porosity may be controlled so that up to 30 per cent by volume of lubricating oil can be retained. As the moulding is performed at high pressures in presses which are largely automatic and have accurately fitted punches and dies, the process shows its economic advantage chiefly in large-quantity production, preferably of standardized sizes, and is best suited to simple cylindrical shapes or bearing disks.

Under light loads, these bearings will operate satisfactorily without

external lubrication, whilst with adequate lubrication they are reported to have been used with pressures as high as 4,000 lb. per sq. in. Their success and their very low oil consumption are probably due to the fact that the supply of oil to the bearing surface is determined by conditions of pressure and temperature and therefore bears a close relation to demand.

Other representative copper alloys used to some extent as bearing metals are here commented on briefly.

Brass: Manganese Bronze. The copper-zinc alloys, chiefly the "60/40" duplex brasses, have been widely used as low-cost bearing materials, despite their poor wear resistance. However, when they are strengthened by small additions of such elements as manganese, aluminium, iron, tin, and nickel, to a total of about 4 per cent, the useful group of "manganese bronzes", with a tensile strength of 30–40 tons per sq. in. approximately, is obtained. The manganese bronzes, like the α-β brasses, have good casting properties, can be readily hot-forged or extruded into tubes and other shapes, and are, therefore, cheap to fabricate; their ingot cost is also low. Where a structural part has to be made in one piece with a bearing bush to save space, these alloys offer good strength with quite good bearing qualities for slow rubbing speeds or oscillating motion. Despite their relatively high strength at normal temperatures, they do not retain their strength at elevated temperatures as well as the aluminium or cast tin-bronzes.

Aluminium Bronze, as used for applications of a bearing character, has a duplex structure, the basic composition being 90 per cent copper, 10 per cent aluminium, and sometimes small additions of iron, manganese, and nickel. It can be sand-cast (recently with additions of up to 9 per cent of lead), extruded and hot-forged, and is an excellent die-casting material. It has a tensile strength of 30–40 tons per sq. in., and a fatigue limit higher than that of any of the alloys so far mentioned ($\pm$14 tons per sq. in. approximately). However, if aluminium bronze bearings are used when rubbing speeds are at all high, excessive wear occurs, which may possibly be due to the presence of an abrasive oxidized film on the bearing surface. Lead additions are said to minimize this effect. Aluminium bronze is used for parts such as wormwheels, gears, feed-screw nuts, and selector forks which mate with hardened steel and which are too highly stressed for phosphor-bronze to be used.

Beryllium Copper usually contains 2·25 per cent beryllium, 0 to 0·35 per cent nickel, and the remainder copper, the tensile strength being 45–80 tons per sq. in. This alloy is expensive, but has been used

TABLE 1. SOME TYPICAL BEARING BRONZES

Type	Nominal composition, per cent					Structure	Approximate average mechanical properties of sand-cast bars			Appropriate standard specification	Description and application
	Copper	Tin	Lead	Zinc	Phosphorus, approx.		Tensile strength, tons per sq. in.	Elongation on 2 inches, per cent	Brinell hardness No.		
Rigid bronzes	85	15	—	—	(0·05)	$\alpha+\delta$	14	2	100	—	Hard wearing bronze, suitable for heavy compressive loads. Used for turntable bearings, locomotive slide valves, etc.
	89	10 min.	—	—	(0·50)	$\alpha+\delta$	19	12 (4 max. chill-cast)	90	2 B 8	Phosphor-bronze, suitable for heavy loads and accurate shaft location, and very widely employed
	88	10	—	2	—	$\alpha+\delta$	19	16	65	B.S.S. No. 383	Admiralty gunmetal. Good casting properties for housings, etc., with integral bearings
Leaded bronze	80	10	10	—	(0·05)	$\alpha+\delta+$ lead	15	12	61	S.A.E. 64	Good anti-friction properties. Applicable where lubrication is somewhat doubtful
Plastic bronze	70	5	25	—	(0·05)	$\alpha+$lead	9	13	42	—	Bronze combining fair strength with considerable plasticity, for use where alignment and lubrication are indifferent
Copper-lead	74	1·2 max.	25	—	—	Copper (or α) +lead	8	15	30	D.T.D. 229	Copper-lead for high-duty aeroplane and automobile crankshaft bearings

successfully in the precipitation-hardened condition for bearings of restricted size subjected to abnormally high pressures, such as thrust bearings in variable-pitch aeroplane propeller bosses, and in bearings for certain instruments.

Silicon Bronzes containing about 3 per cent of silicon with additions of manganese or other elements (tensile strength 20–40 tons per sq. in.) have been used and recommended as bearing bronzes, but little published service information is available, and practical tests have indicated that these alloys, although stronger than the tin bronzes, show some tendency to score the shaft, perhaps due to oxide inclusions.

References

Bassett, H. N. 1937 "Bearing Metals and Alloys," Edward Arnold and Co.

Corse, W. M. 1930 "Bearing Metals and Bearings," Chemical Catalog Company, Inc.

French, H. J., and Staples, E. M. 1929 Proc. Am. Soc. for Testing Materials, vol. 29 (ii), p. 450.

Harder, O. E., and Cole, C. S. 1933 Trans. Am. Foundry Association, vol. 41, p. 314.

Murphy, A. J. 1933 *Metal Industry*, vol. 42, p. 517, etc.

Rolfe, R. T. 1935 *Metal Industry*, vol. 46, p. 691, etc.

INFLUENCE OF PRESSURE ON FILM VISCOSITY IN HEAVILY LOADED BEARINGS

By S. J. Needs, B.S.*

Extensive investigations of heavily loaded journal bearings were conducted in Dr. Kingsbury's laboratory several years ago. The machine used for the present tests was built upon the same principles as those of the original Kingsbury oil-testing machine described previously (Kingsbury 1903). Details of the present machine are necessarily different due to its much greater size.

Theory indicates that with constant film viscosity and speed the friction coefficients of fitted journal bearings and plane surfaces will continue to decrease indefinitely with increasing load. Observations of friction coefficient were found to be in good agreement with theory at loads up to about 1,000 lb. per sq. in. of projected bearing area. At loads of several thousand lb. per sq. in. the friction coefficient curve reached a minimum and then began to increase with added load. In the absence of metallic contact this discrepancy between observed and calculated frictions suggested the probability that the high pressures in the bearing oil film caused an increase in film viscosity. In addition to the extremely low values of the observed friction coefficient, this is one of the most interesting results of the investigations and the present paper undertakes an explanation of this phenomenon.

The most direct analytical approach to the problem seems to be through a consideration of plane surfaces of infinite width. For a first approximation, side leakage may be neglected since the increase in viscosity is a pressure effect and due to pressure alone. When a bearing is operating under steady conditions, side leakage controls the pressure distribution in the oil film, but the actual pressures are due to the external load. Hence, when comparing a mathematical solution for plane surfaces of infinite width with test results of a cylindrical bearing, an error arises from the fact that in an axial cross-section of a practical bearing film the pressures are highest at the centre and fall to atmospheric pressure at the sides. The true pressure effect on viscosity for the film cross-section will be found from the pressure at each point and this will be somewhat different from the approximate effect found from the infinite width assumption of a constant or mean pressure over the cross-section. The curvature of the test bearing surfaces and the approximation involved in the assumed

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law of the variation of viscosity with pressure will also influence the accuracy of the comparison.

The geometry of the plane surfaces is illustrated in Fig. 1. Increasing x is measured positive in the direction opposite motion, hence the linear velocity U of the moving surface is negative.

Tests by Hersey and Shore (1928) and others have shown that the

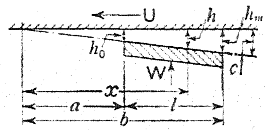

Fig. 1. Geometry of the Plane Surface

logarithm of the absolute viscosity of a lubricant at a given temperature when plotted against pressure, may be approximated by a straight line over a certain pressure range. Fig. 2 shows such a curve and the approximating straight line, the circles representing observations of castor oil at 39 deg. C. Up to 20,000 lb. per sq. in., the straight line gives a fair approximation to the actual conditions for this oil. If r is the slope of the approximating straight line, and the viscosity at

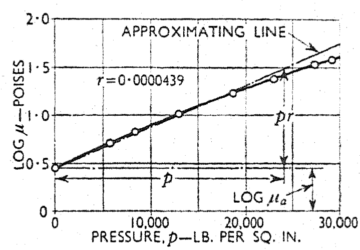

Fig. 2. Viscosity of Castor Oil under Pressure at 39 deg. C. (Hersey and Shore 1928)

atmospheric pressure be denoted by μ_a (μ=coefficient of viscosity), then at any value of p (p=the pressure in the film at any value of x), $\log \mu = \log \mu_a + pr$, from which

$$\mu = \mu_a (10)^{pr} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (1)$$

The differential equation of Reynolds (1886) for the case of infinite width and constant or variable viscosity may be written

$$\frac{d}{dx}\left(\frac{h^3}{12\mu}\frac{dp}{dx}\right) = -\frac{Udh}{2dx} \quad . \; . \; . \; . \; . \; . \; . \; . \quad (2)$$

Substituting (1) in (2), integrating twice and evaluating the integration constants from the boundary conditions $p=0$ when $x=a$ and when $x=b$,

$$\frac{1}{2{\cdot}303r}[1-(10)^{-pr}]=\mathrm{C}\frac{(b-x)(x-a)}{(a+b)x^2}=p_a \quad . \quad . \quad (3)$$

where $\mathrm{C}=6\mu_a\mathrm{U}/c^2$. Writing $s=2{\cdot}303r$

$$p=-\frac{1}{r}\log_{10}(1-sp_a) \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

an expression previously derived by Kingsbury but heretofore unpublished. The load per unit of width is

$$\mathrm{W}=\int_a^b p\,dx=\frac{\mathrm{C}}{\mathrm{D}}\left[\frac{\mathrm{J}}{2}(a+b)-\frac{\mathrm{E}}{s\mathrm{C}}(\mathrm{G}-\mathrm{H})\right] \quad . \quad . \quad . \quad (5)$$

where

$$\left.\begin{array}{l}\mathrm{D}=(a+b+s\mathrm{C}),\quad \mathrm{E}=\sqrt{s\mathrm{C}[4ab(a+b)-s\mathrm{C}(b-a)^2]},\quad \mathrm{J}=\log_e\dfrac{b^2}{a^2}\\[2ex] \mathrm{G}=\tan^{-1}\dfrac{2b(a+b)+s\mathrm{C}(b-a)}{\mathrm{E}},\quad \mathrm{H}=\tan^{-1}\dfrac{2a(a+b)-s\mathrm{C}(b-a)}{\mathrm{E}}\end{array}\right\} . \quad (6)$$

Reynolds's expression for the shearing stress at the moving surface, valid for constant or variable viscosity, is $f=\frac{\mu\mathrm{U}}{h}+\frac{h}{2}\frac{dp}{dx}$, which for the present case reduces to $f=\frac{\mu_a\mathrm{U}}{c}(10)^{pr}\left(\frac{3x_1}{x^2}-\frac{4}{x}\right)$ . . . . . . (7)

where $x_1=2ab/(a+b)$, the value of x for which the pressure is maximum.

The total frictional force per unit width is

$$-\mathrm{F}=\int_a^b f\,dx=\frac{2\mu_a\mathrm{U}}{c\mathrm{D}}\left\{-(a+b)\mathrm{J}+\frac{2}{\mathrm{E}}[3ab\mathrm{D}-s\mathrm{C}(a+b)^2](\mathrm{G}-\mathrm{H})\right\} . \quad (8)$$

and the coefficient of friction is

$$\lambda=\mathrm{F}/\mathrm{W} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Tests with Various Lubricants having the Same Viscosity at the Same Temperatures. The variation of the coefficient of friction with average load is shown in Fig. 3 for a 120 deg. centrally loaded bearing 4 inches in diameter and 4 inches wide. The bearing blocks were of bronze with a $\frac{1}{16}$-inch layer of Babbitt bearing surface. They were carefully fitted to the ground and lapped journal by hand scraping carried

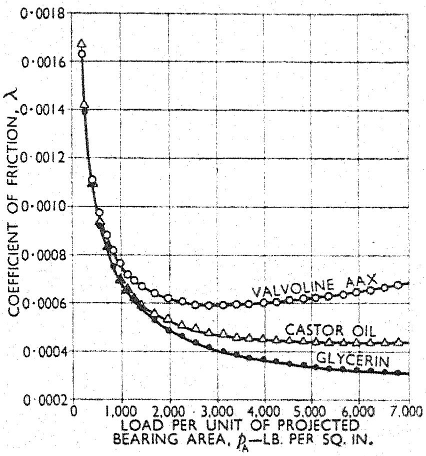

Fig. 3. Variation of the Coefficient of Friction with Average Load

"Valvoline AAX" at 100 deg. F., $\mu_a=37{\cdot}7\times10^{-6}$
Castor oil at 100 deg. F., $\mu_a=38{\cdot}5\times10^{-6}$
Glycerin at 94·5 deg. F., $\mu_a=38{\cdot}5\times10^{-6}$
120 deg. fitted bearing, 4 inches in diameter, 4 inches wide; speed, 11·06 r.p.m. U=2·316 in. per sec.

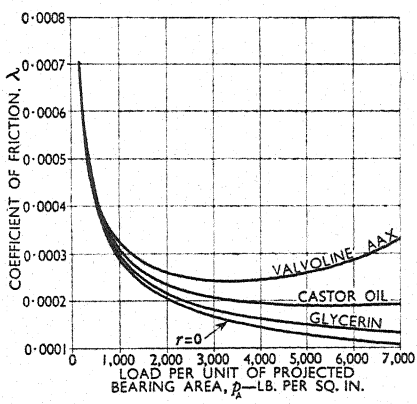

Fig. 4. Variation of the Coefficient of Friction with Load for Plane Surfaces of Infinite Width

"Valvoline AAX" at 100 deg. F., $r=0{\cdot}0000716$.
Castor oil at 100 deg. F., $r=0{\cdot}0000439$.
Glycerin at 94·5 deg. F., $r=0{\cdot}0000172$.
Constant viscosity $\mu_a=38{\cdot}5\times10^{-6}$; $r=0$.
$b=8$ inches, $a=4$ inches; U=2·316 in. per sec.

beyond the point where the journal, lying horizontally in the bearing, would rotate freely at low speed lubricated by atmospheric air. The tests were made at a constant temperature of 100 deg. F. and at 11·06 r.p.m., the linear journal speed being 2·316 in. per sec. Three lubricants having the same viscosity at atmospheric pressure, but having different viscosity-pressure coefficients, are compared. "Valvoline AAX" (a paraffin-base mineral oil) and castor oil have nearly the same viscosity at 100 deg. F. At 94·5 deg. F. glycerin will have the same viscosity as the two oils. With increasing load, the observed curves of the coefficient of friction reach a minimum and then begin to rise. The minimum point was not reached with glycerin, but is quite apparent in the test with mineral oil.

From equations (5), (8), and (9), curves showing the variation of the coefficient of friction with load for plane surfaces of infinite width may be plotted as in Fig. 4, for the same lubricants, temperature and speed as in the bearing tests shown in Fig. 3. The value of r for castor oil was taken from Fig. 2; for glycerin, r was interpolated from data given by Bridgman (1931), and for Valvoline AAX, r was taken from a chart of the Kiesskalt type (1930) prepared by J. F. Spiegel of the Kingsbury Machine Works. Equations for load and friction at constant viscosity ($r=0$) are given by Kingsbury (1932). The values of the coefficient of friction calculated in Fig. 4 are lower than the observed values shown in Fig. 3, since the former are uncorrected for side leakage. It will be noticed, however, that the trend of the calculated curves is the same as that of the observed curves and at the same values of load per unit area the proportionate increase in the coefficient of friction due to the different viscosity-pressure coefficients of the lubricants is approximately the same as calculated. Hence the approximations on which the calculations are based do not appear to introduce any serious errors.

Some unusually low values of friction coefficient are given in Fig. 5. These results were obtained with the bearings used in the tests recorded in Fig. 3. The observations were made at constant speed and temperature. The observed minimum values of friction coefficient and the conditions under which they were obtained are given in Table 1.

Oiliness. Herschel (1922) has defined oiliness as the property that causes a difference in the friction when two lubricants of the same viscosity at the temperature of the film are used under identical conditions. Hersey (1936) has pointed out that Herschel's definition does not specify that the lubricants must have the same viscosity at the pressure of the film and hence the difference in friction between two lubricants having the same viscosity at atmospheric pressure, but

TABLE 1. OBSERVED MINIMUM VALUES OF THE COEFFICIENT OF FRICTION

Lubricant	Minimum coefficient of friction	p_A at min. λ	Speed, r.p.m.	Temperature, deg. F.	μ_a lb. sec. per sq. in.
Olive oil	0·000304	4,250	56·1	120	$4{\cdot}01 \times 10^{-6}$
Castor oil	0·000281	3,800	11·06	135	$13{\cdot}2 \times 10^{-6}$
Glycerin	0·000226	5,500	11·06	120	$14{\cdot}3 \times 10^{-6}$
Kerosene	0·000183	1,500	179·7	81	$0{\cdot}266 \times 10^{-6}$

having different pressure coefficients of viscosity, would be attributed to oiliness.

The conditions under which the tests shown in Fig. 3 were made

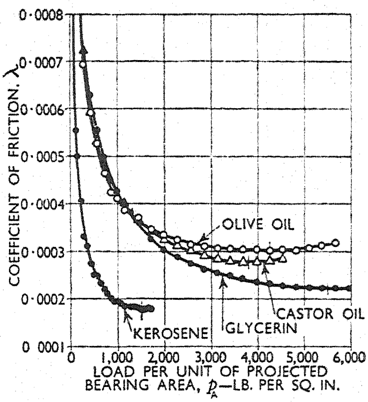

Fig. 5. Unusually Low Values of the Coefficient of Friction

Olive oil at	120 deg. F. and	56·1 r.p.m.	
Castor ,, ,,	135 ,,	,, 11·06 ,,	
Glycerin ,,	120 ,,	,, 11·06 ,,	
Kerosene ,,	81 ,,	,, 179·7 ,,	

120 deg. fitted bearing, 4 inches in diameter, 4 inches wide.

entirely satisfy the requirements of the above definition of oiliness. It is thought that the calculated results (Fig. 4) show satisfactorily that the observed differences in friction are due to the different viscosity-pressure coefficients of the lubricants. From this one might hastily conclude that oiliness is simply a viscosity phenomenon; that the reduction in friction obtained by substituting a vegetable for a mineral

oil is due to nothing more mysterious than the fact that the viscosity of the vegetable oil increases less rapidly than that of the mineral oil under the elevated pressures in the film.

It is known that the friction in a bearing running with a light mineral oil under conditions of incomplete film lubrication, will be reduced by the substitution of a much heavier mineral oil. This reduction in friction is thought to be due to greater separation of the bearing surfaces by the more viscous oil. Another way to reduce the friction would be to replace the light mineral oil with a fatty oil, and then the reduction in friction could be attributed to the lower viscosity-pressure coefficient of the fatty oil. These cases would seem to indicate that oiliness is simply a viscosity phenomenon. But it is equally well known that static friction is reduced by the so-called fatty oils, and under static friction conditions viscosity is thought to be inoperative.

With reference to the true nature of oiliness, the results reported in this paper show that the phenomenon known as oiliness can be produced in bearings by the influence of film pressure on the viscosity of lubricants. Apparently under other conditions, other causes can and do produce the same effects.

References

Bridgman, P. W. 1931 "The Physics of High Pressure." G. Bell and Sons, London.

Herschel, W. H. 1922 Jl. Soc. Automotive Eng., vol. 10, p. 31, "Viscosity and Friction".

Hersey, M. D., and Shore, H. 1928 *Mechanical Engineer*, vol. 50, p. 221, "Viscosity of Lubricants under Pressure".

Hersey, M. D. 1936 "Theory of Lubrication." Chapman and Hall, London; John Wiley and Sons, New York.

Kiesskalt, S. 1930 *Petroleum Zeitschrift*, vol. 26, p. 1, "Die Druckabhängigkeit der Viskosität".

Kingsbury, A. 1903 Trans. A.S.M.E., vol. 24, p. 143, "A New Oil-Testing Machine and Some of its Results".

Kingsbury, A. 1932 Trans. A.S.M.E., vol. 54, RP–54–7, p. 123, "Optimum Conditions in Journal Bearings".

Reynolds, O. 1886 Phil. Trans. Roy. Soc., vol. 177, part 1, p. 157, "On the Theory of Lubrication"; 1901 "Scientific Papers", Macmillan, London, vol. 2, p. 228.

INSTABILITY OF OIL FILMS AND MORE STABLE BEARINGS

By Burt L. Newkirk *

It is known that shafts running on journal bearings at speeds of the order of twice their critical speeds or higher, tend to build up whirling or vibration. This whirling has been called "shaft whipping" and more appropriately "the oil-film whirl" (Newkirk and Taylor 1925). It is not to be confused with instability due to internal friction (Newkirk 1924) in the rotor, which has also been called "whipping" or preferably "cramped shaft whirl." Another form of instability due to action of the oil film has been reported by Stodola (1925) and Hummel (1926). A study of the problem of whirling due to oil film action has been made by Robertson (1933).

The oil-film whirl arises from a potential instability of the oil film which supports a rotating journal. The instability develops into whirling or vibration in co-operation with elastic action of the rotor, elasticity of the supporting structure being also a factor. The development of the whirl from a state of steady running seems to occur as follows: A slight shock causes the rotor to vibrate at its natural frequency, which is numerically equal to the critical or resonant speed. This vibration of the rotor causes periodic displacement of the journal from its equilibrium position on the oil film. When the journal is thus displaced the restoring force that develops in the oil film is not directed exactly toward the equilibrium position. There is a component of the restoring force which sets up a forward whirl of the journal about its equilibrium position. If the speed of the rotor is equal to or greater than some value which is approximately twice the whirl frequency the whirl builds up, otherwise it dies out. The whirl occurs at a resonant frequency of the rotor (usually the lowest resonant frequency) for all speeds of rotation above the lowest speed at which whirling develops.

There is a good deal of variation from this typical behaviour in individual cases. Higher unit bearing loads with correspondingly reduced film thickness tend to raise the speed at or above which instability develops. In some circumstances considerable extension of the range of stable operation can be obtained by decreasing the bearing area and so increasing the unit bearing pressure. The decrease in bearing area is usually effected by shortening the bearing

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and so, because of end leakage, increasing actual maximum unit pressure in even greater proportion. A peripheral groove cut in the middle of the bearing doubles the end leakage area and greatly increases maximum unit pressures with corresponding reduction in film thickness.

In some cases whirling has been overcome by reducing the bearing arc. This also tends to reduce the thickness of the film. On the other hand, elasticity of the supporting structure may reduce the natural shaft vibration or whirl frequency, and the speed at which instability may develop is correspondingly lowered. Internal friction in the bearing supports or other non-rotating parts of the structure may overcome the instability so that the whirl does not build up. The journal diameter, its peripheral speed, and the bearing clearance are factors. The viscosity of the oil has no decisive effect and indications are conflicting. In some carefully conducted laboratory tests stability continued to a higher speed with more viscous oil. Field experience with turbine generator sets gives the opposite indication. In some cases a shock of considerable violence is necessary to start the whirl.

The energy input to the whirl increases at a rapid rate with increasing radius of whirl, but in some cases an equilibrium is reached in which the energy put in by the oil film is absorbed in damping resistance, and there is no further increase in whirl radius. The tendency to instability seems to be more pronounced for rotors having long overhangs such that whirling of the overhanging mass produces abnormally high reaction at the bearings.

Means employed commercially to avoid this instability are :—

(1) Design to keep resonant speed above half running speed.
(2) Higher unit bearing loads.
(3) Special grooving of the bearing surface as shown in Fig. 1 (Newkirk and Grobel 1934).

At the higher peripheral journal speeds (10,000 ft. per min.) unit loads may be 200 lb. per sq. in. or more. Such bearings may have a length less than the journal diameter, with correspondingly high end leakage and high actual maximum unit pressure. Rotors having little or no masses outside the bearings and having unit pressures of this order with short bearings are stable at speeds well above twice their lowest resonant speeds. In the absence of a satisfactory theory to account for this effect it is difficult to determine how much can be accomplished by this means. A model having a journal 2 inches in diameter and a cylindrical bearing $1\frac{1}{8}$ inches long developed instability at 22,000 r.p.m., which was $2\frac{1}{2}$ times the critical speed, with a unit

load (applied through the pull of a magnet) of 133 lb. per sq. in. With a bearing half as long the rotor ran at 30,000 r.p.m. without whirling. The unit pressure was 266 lb. per sq. in. and the speed was $3\frac{1}{2}$ times the critical speed.

The grooving shown in Fig. 1 consists of a comparatively deep (e.g. 0·1 inch) peripheral groove in the lower half of the bearing and a shallow groove (0·020 to 0·070 inch deep) in the upper half, ending in a dam from 45 deg. to 60 deg. from the point of entry of the oil. Lands on the upper half provide bearing area for the journal in case of any tendency to rise against the top of the bearing. At high peripheral speeds (10,000 ft. per min.) pressures of more than 150 lb. per sq. in. develop in the top half of the bearing, the pressure increasing as the speed increases. In general these bearings do not become unstable at any speed at which the rotors can be run. Exception

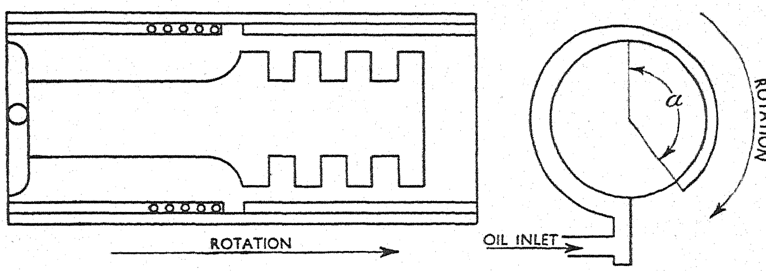

Fig. 1. Bearing with Special Grooving

must be made for rotors having long heavy overhangs and low critical speeds.

The self-imposed load makes these bearings run smoothly where instability could arise on account of low unit load. In bearings for gear pinions, for example, the bearing load may reverse its direction as the load transmitted through the gear increases. The vibration which would develop in the transition range is avoided by using this design. Three-bearing turbine generator sets frequently have one bearing which must carry a low unit load and in this case also bearings of this design give more satisfactory performance.

This instability of the oil film seems likely to become increasingly important and a satisfactory hydrodynamic theory to account for it is needed. Stated briefly the phenomenon to be explained is : When a journal running on an oil film is caused to travel (whirl) about its position of equilibrium, the film exerts a reaction component on the journal tangent to the path. The space integral of these components, summed for a complete circuit, is positive when the rotational speed

of the journal is of the order of twice the whirl speed, or higher. The value of this integral increases with increasing amplitude of whirl.

References

Hummel, C. 1926 V.D.I. Forschungsarbeiten, No. 287, " Kritische Drehzahlen als Folge der Nachgiebigkeit des Schmiermittels im Lager".

Newkirk, B. L. 1924 *General Electric Review*, March, p. 169, "Shaft Whipping."

Newkirk, B. L., and Taylor, H. D. 1925 *General Electric Review*, August, p. 559, "Shaft Whipping due to Oil Action in Journal Bearings".

Newkirk, B. L., and Grobel, L. P. 1934 Trans. A.S.M.E., vol. 56, APM–56–10, p. 607, " Oil-film Whirl—A Non-whirling Bearing".

Robertson, D. 1933 *Philosophical Magazine*, series 7, vol. 15, p. 113, "Whirling of a Journal in a Sleeve Bearing".

Stodola, A. 1925 *Schweizerische Bauzeitung*, vol. 85, p. 265, "Kritische Wellenstörungen infolge Nachgiebigkeit des Ölpolsters im Lager".

THE NOMY PAD BEARING: DEVELOPMENT AND DESIGN

By Professor F. K. G. Odqvist*

Since the days of Osborne Reynolds, many efforts have been made to realize ideal conditions of fluid film lubrication in bearings. The principal deduction from Reynolds's theory was the need to arrange for a wedge-shaped oil film to be formed between the metal surfaces, narrowing in the direction of motion, should the internal friction in the fluid be capable of transmitting pressure through the oil film.

Among those efforts, one may mention the bearings of A. Kingsbury and A. G. M. Michell. The designers of these bearings have taken up Osborne Reynolds's idea of splitting up the bearing surface into a series of pads, each one eccentrically supported and capable of individual tilting, whereby an increased load-carrying capacity is ensured. The application of the Kingsbury or Michell design has been mainly to thrust bearings, the corresponding journal bearings forming a very limited field.†

Universal application of the idea of the tilting pad has been prevented because the eccentric support of the pads necessitates the use of one direction of rotation only. For the opposite direction, the pad theoretically acts merely as an oil scraper. Thus, the invention of the tilting pad must be completed with a device which will enable the pads to change their point of support when the direction of rotation of the journal is reversed.

The Nomy pad bearing (Fig. 1) represents an effort to realize this intention by making the pads rotate with the shaft and take up different positions in relation to certain abutments according to the direction of rotation. The abutments project from an inner bearing ring shrunk on to the shaft and forming supports for the pads.

For a journal bearing, this is shown diagrammatically in Fig. 2. The corresponding feature of a Nomy single-acting thrust bearing is shown in Fig. 3. The inner ring A has two supporting ridges G for each pad B, one ridge for each direction of rotation. The abutments are shown at F. The journal bearing comprises inner and outer ring, pads, holders, caps and oil throwers, and forms a complete unit (Fig. 4). Both sliding surfaces are finished by the manufacturer. The user is responsible only for the shaft, which has

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† Cf. Michell, A. G. M. Trans. A.S.M.E., 1929, vol. 51, MSP–51–21.

to be ground within ordinary limits for ball bearing mounting. Mounting with a split cone-sleeve is also possible. The outer dimensions D and B (Fig. 4) are the same as those of standard ball bearings.

On reversing the direction of rotation of a Nomy pad bearing, the

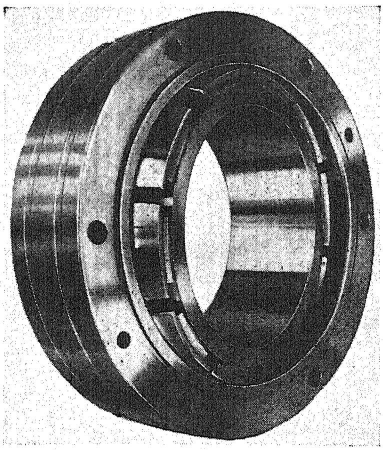

Fig. 1. Nomy Pad Journal Bearing

pads will adjust themselves to the correct position in the unloaded area of the periphery. Such unloaded areas of the periphery will always exist in a journal bearing having its principal load in one constant direction. In a thrust bearing such an unloaded area may be

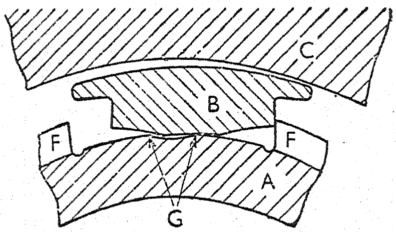

Fig. 2. Arrangement of Inner Ring A, with Abutments F and Pad B

formed artificially, for instance, by having a wedge-shaped recess of a length little more than one pad, measured in the direction of motion (Fig. 5). To avoid edge pressures due to misalignment of the journal the pads and sliding surfaces are spherically shaped.

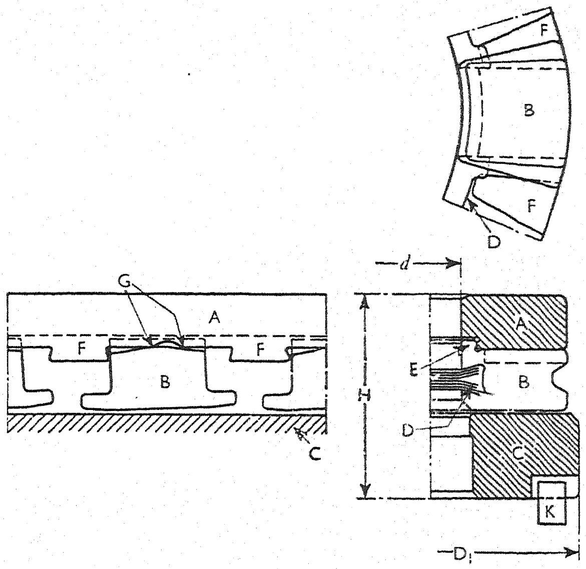

Fig. 3. Single-Acting Thrust Block Bearing

A Inner ring. B Pad. C Sliding ring. D Pad holder. E Radial flange. F Abutment. G Supporting ridge. K Locking pin.

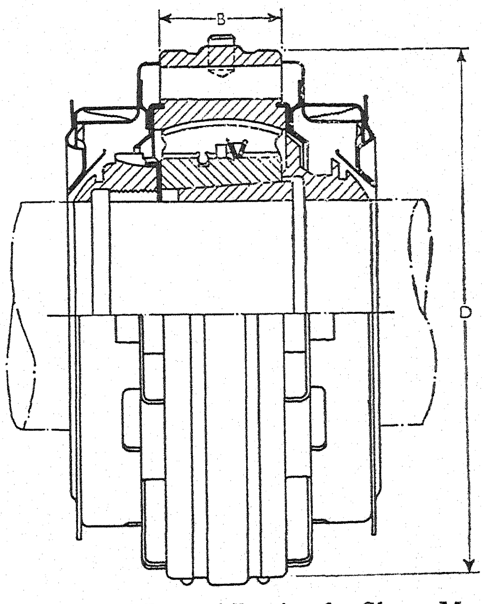

Fig. 4. Nomy Pad Journal Bearing for Sleeve Mounting

The arrangement with rotating pads in a journal bearing has a noticeable advantage over that with stationary pads from a theoretical point of view. Fig. 6 shows the action of forces P_1, which would be generated according to Reynolds's theory in a bearing with six stationary

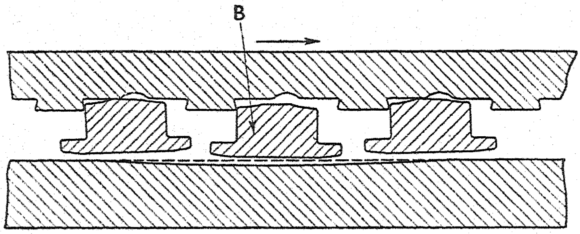

Fig. 5. Artificial Unloaded Area at Periphery in Thrust Bearing

pads due to the translational motion of the surface of the journal, the internal motion of the fluid being truly stationary. The journal centre will be displaced in the direction of the load Q, which is equal and opposite to the resultant R_1, of the forces P_1. Fig. 7 illustrates the

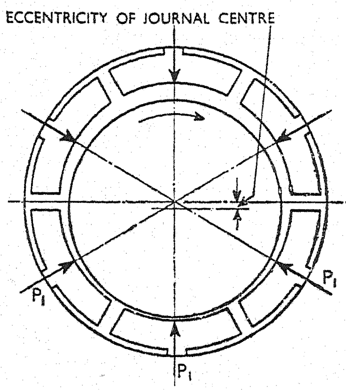

Fig. 6. Distribution of Pressure on Stationary Pads

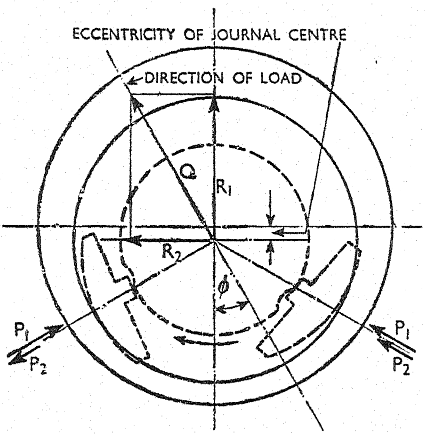

Fig. 7. Distribution of Pressure on Rotating Pads

corresponding action with rotating pads and a stationary outer sliding surface. Speed, clearance, and viscosity being constant, the translational motion of the pads gives rise to exactly the same forces P_1 across the oil film. In addition to these, however, other forces P_2 occur because the motion of the pads has now a pulsating character, each pad ap-

proaching the outer sliding surface during one half-revolution and leaving it during the other half. The forces P_2 will have a resultant R_2 at right-angles to the resultant R_1 of the forces P_1, as mentioned before. R_2 will combine with R_1 to neutralize the bearing load Q, which accordingly will be increased in the proportion $1/\cos\phi$. Here the journal centre will be displaced in a radial direction as much as with stationary pads and, in addition, will attain an angular displacement ϕ from the direction of the load.

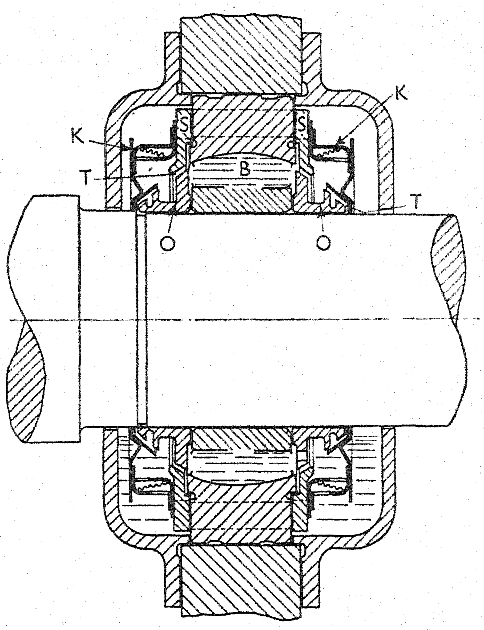

Fig. 8. Method of Lubrication and Sealing in the Nomy Journal Bearing

In a radial bearing, the arrangement with rotating pads has a decided advantage in ease of lubrication. Each pad takes up a sip of oil when passing through the oil reservoir at the bottom of the bearing. The oil is held against the outer ring by centrifugal force. Thus the oil fills the spherical space between the pads, which travel entirely submerged in a rotating oil bath B (Fig. 8). This spherical space is extended laterally by baffles S attached to the outer ring. To facilitate the entrance of the oil, particularly at the start, rotating oil throwers O are fixed on to the shaft on either side of the bearing, forcing the oil into the spherical space B. To prevent the oil from being thrown

round in the bearing housing, the baffles are extended to caps K forming approximately spherical sealing slots T with the corresponding part of the oil thrower O. These slots maintain a small constant clearance which is independent of the position of the journal within a misalignment of 1/200. The journal keeps dry outside the slot T, which accordingly acts as a sealing device for the oil. At high speed the oil will be mixed with air bubbles. These bubbles and the foam

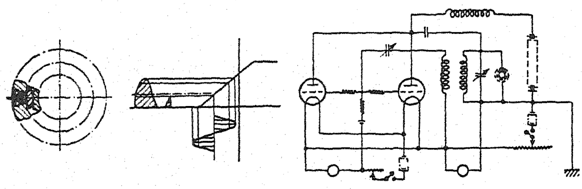

Fig. 9. Ultramicrometer Circuit for Measuring Oil Film Thickness

generated are kept inside the cap K, and the oil level of the bearing housing remains perfectly quiet outside. In the rotating oil space B between the pads, the air bubbles are separated in the direction of the journal and clean oil is supplied to the sliding surface.

These characteristics of the Nomy journal bearing can be verified in laboratory tests. Thus the formation of a wedge-shaped oil film between pad and outer ring, and more particularly the thickness of

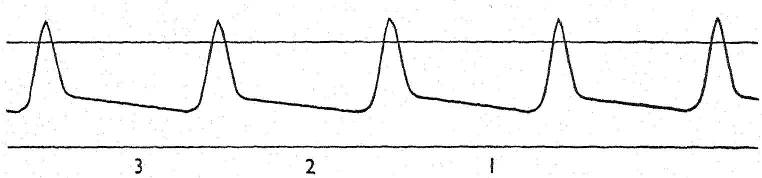

Fig. 10. Oscillogram showing Variations in Film Thickness in a Bearing Running under Load

this film, may be checked and measured in a bearing running under load, using an electrical capacity method. Condensers are inserted in the outer bearing ring and their capacity in relation to the rotating pads may be measured, the latter being earthed through the journal. Thus, a high-frequency current is generated in a thermionic valve circuit, the diagram of which is seen in Fig. 9, forming a development of the Dowling "ultramicrometer".* The only important

* Dowling, Proc. Roy. Dublin Soc., 1921 vol. 16, p. 18; *Engineering*, 1921, vol. 112, p. 395. For the development of the circuit of Fig. 9, cf. Ekelöf, S. *Teknisk Tidskrift, Mekanik*, 1929, 21st Sept.

points are that the current in the circuit should be very sensitive to variations in capacity (about 1–4 milliamperes per microfarad) and that the capacity should be read at a constant mean value of the current.

In Fig. 10 is reproduced an oscillogram of the current, e.g. of the capacity variations in one of the condensers. The wedge shape of the oil film is easily visible, the blocks apparently moving from right to left. The cusps correspond to the space between the blocks.

FILM LUBRICATION APPLIED TO RAILWAY AXLE BEARINGS

By J. Foster Petree, M.I.Mech.E.*

Most papers dealing with the design of journal bearings pay a tribute to "the classical work of Beauchamp Tower," and the majority, in so doing, are referring to the First Report on Friction Experiments (1883). The experiments described in that report related to a steel journal, 4 inches in diameter and 6 inches long, having a gunmetal brass "embracing somewhat less than half the circumference of the journal, on its upper side", and lubricated by dipping into a bath of oil. In dimensions and design, this experimental bearing represents the simplest form of railway axle bearing; in fact, bearings of this type, resting on the journal and loaded from above, are seldom encountered elsewhere than on railway axles and in rolling mills. In spite of the remarkable results obtained by Tower, however, and the revolutionary developments in bearing design subsequently deriving from them, the railway axle bearing, in general, has profited less from Tower's work than almost any other class of bearing commonly employed. Until recently it continued to rely on grease rather than oil as the lubricant, and even in oil-lubricated bearings the principle of the oil bath, feeding the bearing with as much oil as it can use, has been disregarded in favour of the oil pad, which can rarely provide sufficient oil to form a true fluid film.

A major difficulty in the design of railway axle bearings is that much more is required of the bearing than merely to support a steady vertical load. The bearing surface must extend downward nearly to the horizontal centre line of the axle in order to act as a thrust bearing; and this applies, whether the bearing is on a locomotive axle, through which the thrust is exerted, or a train axle, through which it is received. Accelerations and decelerations cause varying loads in a horizontal direction and, at the same time, the action of the springs and the effect of track irregularities, such as points, crossings, and rail joints, produce fluctuating vertical loads and also shock loads acting at rapidly changing angular positions between the horizontal and the vertical. In addition, there are end shocks on the brasses due to the side play required to enable the vehicle to negotiate curves. It is significant that much bearing deterioration begins by the breaking away of whitemetal at the ends of the brasses.

Osborne Reynolds showed (1886), and other investigators have

* Consulting engineer.

since confirmed, that, for the maintenance of a sufficient oil film, the journal and the brass must be free to take up the correct relative position corresponding to the load, speed, and viscosity of the lubricant. If this requirement is satisfied the film will be permanent enough to resist breakdown by variation of load; but if the relative freedom is such that the brass can be suddenly detached from the journal, rupture of the film is inevitable, and most of the oil in it will flow away before the brass returns to its bed on the journal. Therefore the relative movement must be restrained so that the clearance cannot be suddenly increased beyond the ability of the oil to flow in and fill the space. This ability to meet a suddenly increased oil demand is negligible in a bearing lubricated only by the capillary action of an oil-soaked pad, and requires that the bearing must be flooded with oil to an extent far beyond the supply needed for steady running.

At this point it is convenient to summarize the factors essential to an axle bearing in which a permanent oil film is to be maintained. It must have:—

(1) Sufficient area to support the direct load.
(2) Sufficient area suitably placed to withstand horizontal shocks and the overloads due to braking, etc.
(3) Strength to resist distortion under the foregoing loads, or under end shocks due to side play.
(4) Ample oil supply for any emergency.
(5) Proportions and clearances that will ensure a correct oil film under all conditions.
(6) An assured formation of the film immediately relative movement takes place between journal and brass.

The above points represent essentials, but the following features are also highly desirable:—

(7) The bearing should be unaffected by climatic conditions.
(8) It should be completely sealed to avoid leakage and to prevent contamination or theft of the oil.
(9) It should require no attention on the road for as long as the wheel set of which it is a part remains in continuous service (this should follow naturally, if Nos. 7 and 8 are satisfied).
(10) So that the film-lubricated box can be interchangeable with older types, the design should necessitate no alteration to the form or dimensions of the standard plain axle.

If the design and the standard of construction conform with the above requirements, more especially with Nos. 9 and 10, the reduction in maintenance cost will more than compensate for a considerable increase in first cost as compared with that of the simple pad-lubricated

box, which appears to be the general cost criterion. Low first cost is of greatest importance where long continuous hauls are relatively infrequent and railway wagons are used more for storage than for transport, the importance diminishing with more intensive use. For passenger stock, especially on suburban services, now usually operated with set trains, a high availability ratio is a prime consideration, and a bearing that consistently meets condition No. 9 is justified almost irrespective of cost, within the limits likely to be encountered, keeping in mind the useful life of the stock.

The experiments of Boswall and Brierley (1932) have shown that the most satisfactory film is obtained with bedded brasses when the bearing surface has an arc of contact of between 45 deg. and 60 deg., and in

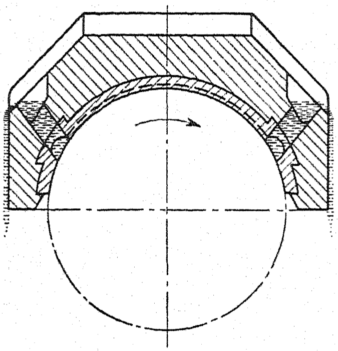

Fig. 1. The Isothermos Bearing

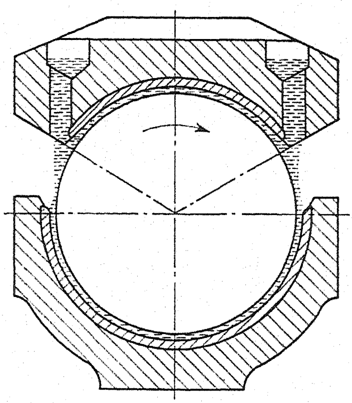

Fig. 2. The Peyinghaus Axle Bearing

this respect the standard Railway Clearing House boxes, which are pad-lubricated, are well designed; but as the quantity of oil is insufficient to guarantee a full film, and no assurance is possible that the brass and the journal are maintaining the correct relative attitude, it is doubtful whether, in practice, true fluid film lubrication is ever obtained in them otherwise than fortuitously, and then only for a very brief space of time.

Various patents have been taken out for bearings embodying oil-circulating devices, such as lifters or palettes attached to the end of the axle and revolving with it, which pick up oil from a sump and deliver it to troughs formed in the top of the brass, whence it flows by gravity through cast or drilled holes on to the surface of the journal. Some of these patented bearings are in regular use, particularly on the

Continent, and have demonstrated that this method of oil circulation is quite effective. Some particulars of one of these, the Isothermos bearing (Fig. 1), are quoted by Haslegrave (1935). Another is the Iracier box of Wood and Carson (Ahrons 1926). None of them, however, entirely solves the problem of preventing "jump"; and in reducing the bearing arc to suitable fluid-film proportions, the resistance to side and end shocks is not always sufficiently maintained. Moreover, in single-brass bearings of this type, there is an inevitable time lag, after

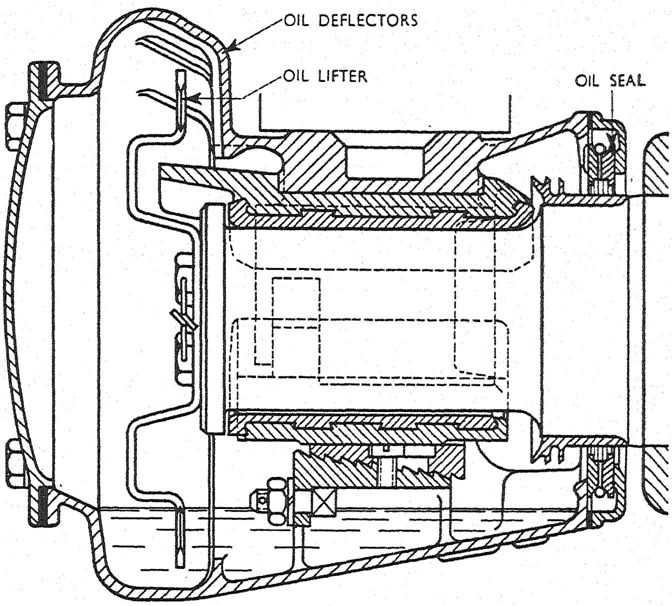

Fig. 3. Section of a Peyinghaus Axlebox

prolonged standing, before the oil picked up by the lifter reaches the actual bearing surface.

A development in oil circulation, providing a restraint of excessive journal movement while restoring the advantages of true bath lubrication as applied by Tower, has been evolved in Germany in the Peyinghaus axle bearing (Fig. 2), probably the most advanced design at present in general use. In this an under-brass is fitted, clear of the journal but close enough to retain a reserve of oil when standing, which is carried to the upper brass immediately movement takes place; and serving also to limit separation of the journal and the upper brass when crossing points or rail joints. As the bearing is completely sealed, wagons fitted with it can be inverted in tipplers; a consideration

which, in the past, has undoubtedly contributed to the retention of grease-lubricated boxes in coal wagons. The section (Fig. 3) is of a Peyinghaus axlebox used on the German State Railways, and having a vertical adjustment to the under-brass, the specification requiring that the boxes must be interchangeable on journals differing in diameter by a maximum of 6 mm. They are supplied under guarantee to run without replenishment or other attention until the wheel set is taken out, in the ordinary course, for tyre-turning. Some test results in the makers' shop and in service, with loads up to 12 metric tons per bearing, are given (Fig. 4). Steelworks wagons carrying 70-ton ladles are also in use, loaded to 25 metric tons per bearing.

Although experience with these bearings has been satisfactory under

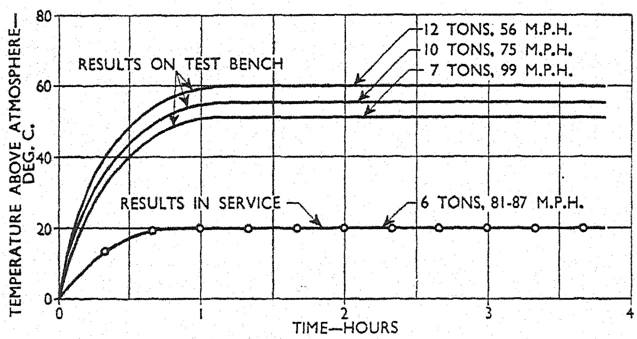

Fig. 4. Test Bench and Service Results

climatic conditions ranging in location from South America to the Arctic Circle, improvement appeared possible in the arrangements for dealing with horizontal loads, especially those due to shunting and braking, and tests were made at the National Physical Laboratory with multiple-brass bearings designed to oppose the cushioning effect of a full oil film to any load likely to be encountered in service, whatever its direction of application around the journal periphery. A similar bearing with a four-part surface was tested at the Technische Hochschule, Berlin, mounted in a locomotive inner axlebox on a journal 110 mm. in diameter and 187 mm. in effective length, supported between two roller bearings and run at speeds up to 1,200 r.p.m. with loads up to 4,000 kg., and at speeds up to 532 r.p.m. with loads up to 10,000 kg. At the latter loading the coefficient of friction was 0·0012 at 372 r.p.m. and 0·0020 at 532 r.p.m. The bearing was lubricated solely by its own self-contained oil supply and functioned satisfactorily throughout, but the initial tests were twice interrupted by failure of the roller bearings, although each of these carried only half of the test load.

The National Physical Laboratory tests were made with two-, three-, and four-part brasses of 134 mm. effective length, fitted in a spherical holder and run on a journal 100 mm. in diameter at speeds up to 1,500 r.p.m. and loads up to 6 tons—the limit of the apparatus. To test the centring effect of the several separate oil films additional tests were made at speeds up to 800 r.p.m., when a definite centring influence was observed which was little affected by the loading. By permission of the Director of the National Physical Laboratory, curves are

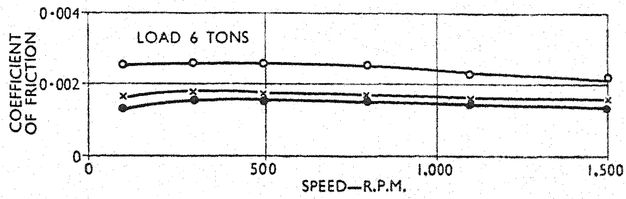

Fig. 5. Variation of the Coefficient of Friction with Speed
● Two-part brasses. × Three-part brasses. ○ Four-part brasses.

appended (Fig. 5) showing the variation of friction for these bearings, together with some conclusions from the report, and dial gauge readings of the shift of the journal in the four-part bearing at speeds of 100, 500, and 800 r.p.m., when carrying loads of 2 tons, 4 tons, and 6 tons. No bearings of this type have yet been tried in railway service, but the indications are that the design possesses sufficient merit to justify experiment on a wider scale.

APPENDIX

Table 1. Extracts from National Physical Laboratory Report on Tests of Three Multiple-Brass Bearings

III. *Horizontal Shift and Vertical Lift.* (*c*) Four-Brass Bearing

One unit (1·0)=0·001 inch

Load . . .	2 tons			4 tons			6 tons		
Speed, r.p.m. . .	100	500	800	100	500	800	100	500	800
Horizontal shift .	1·0	1·3	1·7	0·5	0·8	1·0	0·4	0·5	0·7
Vertical lift . .	0·7	1·2	1·6	0·7	1·1	1·3	0·7	0·9	1·0
Temperature, deg. C.	25·0	43·0	45·0	29·5	43·5	45·5	23·5	44·0	51·0
Minimum film thickness . . .	0·6	1·0	1·3	0·7	1·0	1·2	0·7	0·9	1·0

Discussion of Observations

Measurements of Horizontal Shift and Vertical Lift:—

(*a*) The observations showed that the horizontal shift and the vertical lift increased with the speed and decreased with the load.

(*b*) The minimum film thickness increased with increase of speed and diminished slightly with increase in load.

(*c*) The increase in the number of brasses caused a decrease in the horizontal shift and an increase in the minimum film thickness.

Conclusions. The bearings ran satisfactorily throughout the tests with no measurable wear, small friction loss, and apparently under conditions of true fluid friction. The tests showed that by increasing the number of brasses the horizontal shift was decreased and the film thickness increased under any specified conditions of load and speed.

References

AHRONS, E. L. 1926 "Lubrication of Locomotives", Locomotive Publishing Company, London.

BOSWALL, R. O., and BRIERLEY, J. C. 1932 Proc. I.Mech.E., vol. 122, p. 507.

HASLEGRAVE, H. L. 1935 Proc. I.Mech.E., vol. 129, p. 470.

REYNOLDS, O. 1886 Phil. Trans. Roy. Soc., part 1, p 157.

TOWER, B. 1883 Proc. I.Mech.E., p. 632.

A RE-EXAMINATION OF THE HYDRODYNAMIC THEORY OF BEARING LUBRICATION

By Professor L. Prandtl *

Though the hydrodynamic theory of bearing lubrication has been widely accepted, it has also met with criticism and even those who accept the theory consider that the conditions under which it leads to reliable conclusions require further clarification. For this reason a machine (Fig. 1) was built to enable experiments to be made on lubricated bearing brasses under particularly clear conditions. By means of levers 1 and 2, which constitute a pair of tongs, three brasses 3, with ball joints, are mounted on a polished steel shaft 4, driven by a variable-speed motor. The bearing is loaded by means of an adjustable

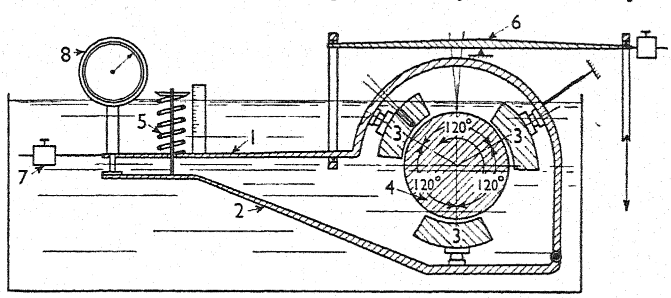

Fig. 1. Bearing-Testing Apparatus

spring 5. The levers 1 and 2 are suspended from a lever 6, the other end of which carries a scale pan. A movable weight 7 is provided to balance the unloaded tongs. As the three loading forces on the brasses due to the tongs form 120 deg. angles and are all equal, application of the principle of virtual displacement shows that the movement of levers 1 and 2, which can be read on dial 8, is proportional to the total radial movement of the three brasses. Torque due to friction results in a load on the lever 6, which can be measured by adding weights on the scale pan. The load on the tongs is very small compared with that on the brasses, so that the resulting difference in pressure can be neglected. The bearing brasses, which so far have had an axial width of 60 mm. and a circumferential length of 20 mm., are, as stated, free to move on their ball joints. Owing to the pressure distribution in the oil wedge they automatically assume a position

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parallel to the axis. The first experiments were made with brasses which, when stationary, fitted perfectly on the shaft, i.e. their inside radius was the same as that of the shaft. Thus they correspond to the sliding pads of a Michell bearing, the theory of which has been particularly well worked out.*

In the tests, carried out by Frössel, the oil supplied in abundance to the three brasses was found to gather into a thick swelling before each brass. Nevertheless the brasses lifted but little from the shaft and on the whole it was impossible to get the same readings from one day to another. Further, the figures obtained did not agree with theory. It was then decided to submerge the whole of the rubbing parts in oil. The brasses had to be lifted from the shaft at starting, but they then took up position accurately, so that readings taken at intervals of months always came out the same, just as if they had been taken in the same hour. The behaviour now corresponded in every detail to theoretical expectations.

The following notation is used: η is the viscosity of the oil; u is the peripheral speed of the journal, l the circumferential length of the brass, h_m the mean thickness of the oil film, p_m the mean pressure in the oil film, and τ_m the average shearing stress on the surface of the journal. Then if, on the one hand, some of the constant geometrical data are suppressed,

$$\tau_m \propto \eta u/h_m \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

On the other hand, the pressure rise in the direction of flow, dp/dx, which can be expressed as p_m/l by suppressing numerical factors, can be connected as follows with the shearing stress τ_m:—

$$p_m/l \propto \tau_m/h_m \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

By eliminating τ_m from equations (1) and (2),

$$p_m/l \propto \eta u/h_m^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

so that

$$h_m \propto \sqrt{\eta u l/p_m} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

By introducing the load intensity per axial unit length of the brasses, $P=p_m l$, then $\eta u/p_m l$ can be written $\eta u/P$, a dimensionless quantity, which is characteristic for the bearing, and may be written shortly as B. Equation (1) then becomes

$$h_m \propto l\sqrt{\eta u/P}=l\sqrt{B} \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

* Cf. Prandtl, L. "Abriss der Strömungslehre", p. 109, or "The Physics of Solids and Fluids", p. 292 (1st ed.). The problem was first treated by Osborne Reynolds, Phil. Trans. Roy. Soc., 1886, part 1, or "Papers", vol. 2, p. 228.

From equation (1)

$$\tau_m \propto \eta u/l\sqrt{B} \propto 1/l\sqrt{\eta u P} = P/l\sqrt{B} \quad . \quad . \quad . \quad . \quad . \quad (6)$$

For a proof of equations (5) and (6) h_m or τ_m can be plotted against the

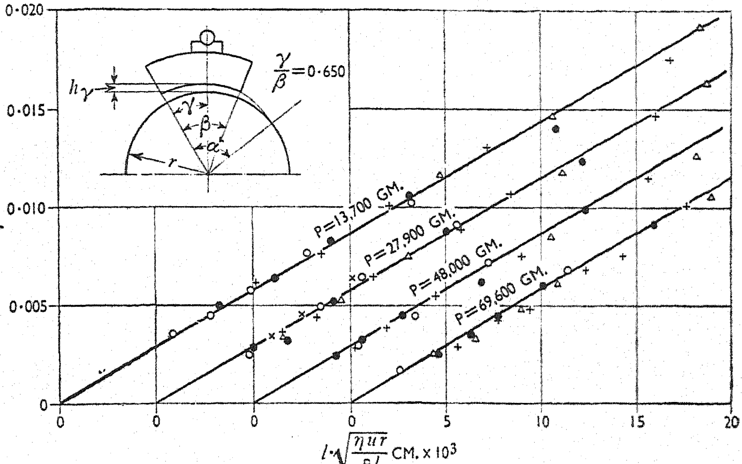

Fig. 2. Film Thickness h_γ depending on $l\sqrt{Br/l}$*

×44 ○ 88 ● 176 + 264 △ 352 cm. per sec.

expressions on the right-hand side of the equation so that the relationship must be linear (Fig. 2 and 3). In place of the mean thickness h_m,

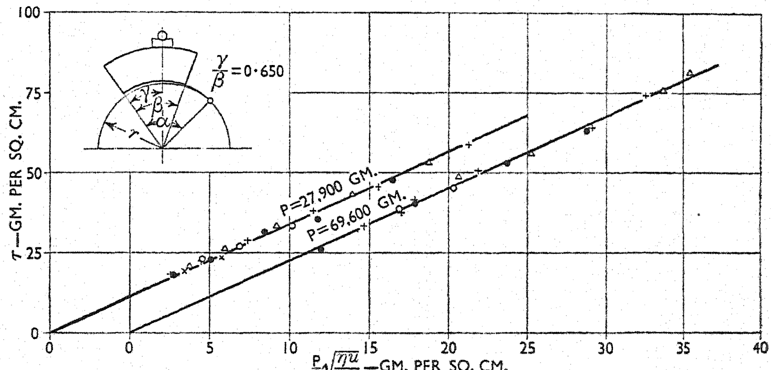

Fig. 3. Shearing Stress τ depending on $P/l\sqrt{B}$

×44 ○ 88 ● 176 + 264 △ 352 cm. per sec.

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* The values of P in Figs. 2–4 indicate the total load on one of the brasses and not the specific load P of the formulæ.

the measurable variation in distance between journal and bearing-brass joint is represented by h_γ. The factor $\sqrt{l/r}$ employed in Fig. 2 and also in Fig. 4, which are taken from another publication, can be neglected. Individual readings correspond to different loads, different peripheral velocities and also different oil viscosities. It will be seen that straight-line relationship is well maintained and that very small thicknesses—varying from 0·02 to 0·2 mm.—of the oil film can be measured satisfactorily.

The Michell theory requires that at a constant centre of pressure (0·65l in the experiments), the lift of the front edge and rear edge should remain in the same relation. Therefore, the angle of tilt of the pad must be directly proportional to h_m/l, so that

$$h_m/l \propto \sqrt{B} = \sqrt{\eta u/P}$$

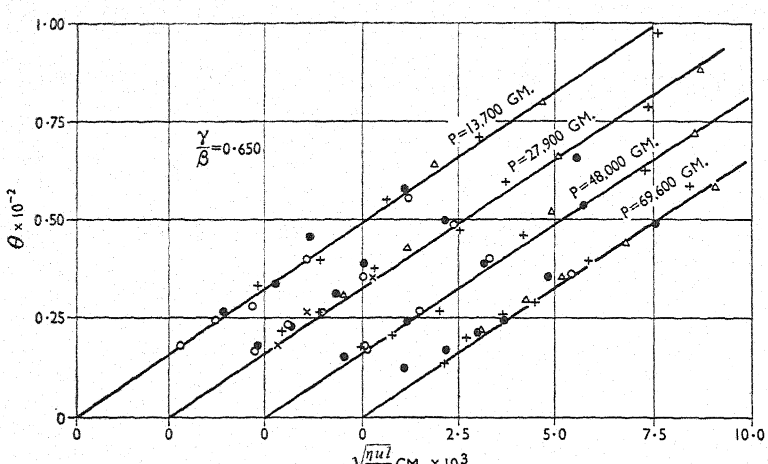

Fig. 4. Distortion of the Bearing Brasses
× 44 ○ 88 ● 176 + 264 △ 352 cm. per sec.

The distortion of the bearing brasses in the apparatus, as compared with the stationary position, was measured by means of a mirror attached to one of the brasses and observed through a telescope fitted with a scale. Fig. 4 records some such measurements and the proportionality is again seen to be good. One of the brasses was provided with holes which were connected to a pressure gauge. The pressures recorded (Fig. 5) not only gave the exact picture which would have been expected from theory, but were quite repeatable.

The conclusion is drawn that the hydrodynamic theory is correct in every respect. Deviations are to be expected with higher loads,

under which the oil temperature due to frictional heat increases substantially from the front to rear edge of the brasses, so that the viscosity cannot be considered to be constant over the whole area. For the loads employed in the experiments, the oil temperature, measured by means of a thermocouple (Fig. 1), should be much the same as that of the brasses.

The following explanation of the unsatisfactory action of the research apparatus, in which the oil was only thrown on to the journal, is put forward. The oil film carried by the journal from one brass to another apparently became covered with a film of air which was not dispersed by the incoming oil but was drawn into the bearing clearance by the rotation of the journal. The expression $\tau_m h_m/\eta u$ (cf. equation (1)), which should according to theory be somewhat greater than unity, has been derived for the values of τ_m measured under these service conditions and for the estimated values of h_m, which lie at the lower limit of experimental accuracy. As the value of the expression was found to be considerably less than unity in the tests, some form of "air lubrication" must be assumed, one oil film sliding over the other in such a way that the air film acts as a lubricant. The local reduction in viscosity due to a rise in temperature could also play a part. However, the values obtained for the foregoing expression are so small that such an explanation must be rejected. The presence of the film of air provides an explanation why hydrodynamic theory does not apply in this case. It is probable that such a penetration of a thin layer of air may also occur in commercial bearings.

Further tests were carried out with brasses whose radius was 1 mm. larger than that of the journal. This gives conditions for the oil wedge which are similar to those obtaining when a cylindrical pad is moved over a flat surface, the radius of curvature being

$$r' = \frac{1}{1/r_2 - 1/r_1} = \frac{r_1 r_2}{r_1 - r_2} \quad . \quad . \quad . \quad . \quad . \quad (7)$$

where r_1 and r_2 are the radii of the bearing brass and journal respectively. These experiments should throw light on conditions in an ordinary bearing with a large clearance. In this case variation of the load on the brasses does not result as before in similar distributions of the pressure, but the pressure zone decreases with decrease in the oil film thickness h_0. This can also be seen from a geometrical consideration. The distribution of the pressure depends on the ratio between the thickness of the oil film in one particular place and the smallest thickness of the oil film. Obviously, a characteristic length for the pressure film zone is the distance between the place with the smallest oil thickness and that where the thickness is twice as great. Simple geometrical considerations provide a value for such a distance

of $\sqrt{2r'h_0}$, abbreviated as l_0, which, in applying the aforementioned dimensions, will play the same part as the length of the bearing brass l. This holds good only when l_0 is small compared with the actual length of the bearing brass, as only in this case is the pressure zone. practically confined to the middle area of the length of the brass. When this condition is fulfilled, equations (5) and (6) can be written

$$h_0 \propto l_0\sqrt{\eta u/\mathrm{P}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5a)$$

and

$$\mathrm{T} \propto l_0 \tau_m \propto \mathrm{P}\sqrt{\eta u/\mathrm{P}} \quad . \quad . \quad . \quad . \quad . \quad (6a)$$

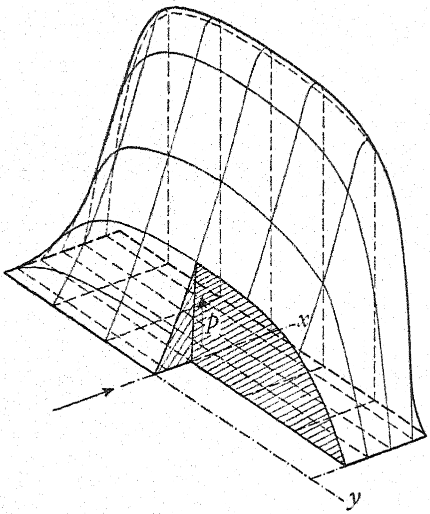

Fig. 5. Pressure Distribution for $r_2 = r_1$

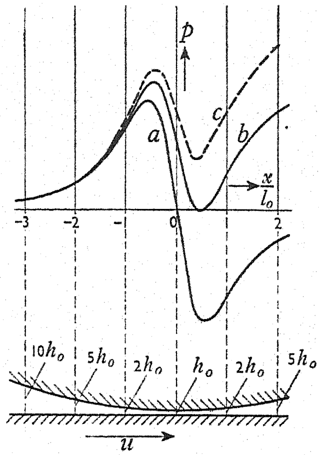

Fig. 6. Theoretical Pressure Distribution for $r_2 > r_1$

The applicability of equation (6a), where T represents the resultant circumferential stress, is obvious. Equation (5a), on the other hand, when l_0 is replaced by its equivalent $\sqrt{2r'h_0}$, still contains h_0 on the right-hand side. On squaring and abbreviating this equation then

$$h_0 \propto r'\eta u/\mathrm{P} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

which is sufficiently confirmed by the experiments under the above-mentioned restrictions.

The pressure distribution also depends, with the same restrictions, on the length l_0, which means that practically similar curves are obtained under different loads and speeds when the measured pressures are made dimensionless with the pressure P/l_0, and are plotted on the abscissa x/l_0 (Fig. 7). The deviation of the single curves of

Fig. 7 at the lower left-hand side is due to the finite length of the brass which, according to the experiment, appears longer or shorter, depending on its relation to the length l_0. In Fig. 7 the calculated pressure distribution is represented by a dotted curve. The deviation from the observed curves is due to the fact that the calculation is made for two-dimensional flow, therefore for a bearing of infinite axial length.

From the theoretical point of view the following question is not yet settled and should, therefore, be cleared up by these experiments. When for such a bearing it is assumed that the position of the narrowest oil wedge is exactly in the middle, so that the wedges are equally wide at both ends and that the pressure at both ends is equal to atmospheric pressure, then the pressure distribution will be as in Fig. 6, curve *a*, which obviously gives a resulting force equal to zero. To the high pressures in front of the narrowest areas correspond the the equally high under-pressures behind those areas. From physical considerations it would be expected that, as the oil is never entirely free from air, the under-pressure would at most approach the absolute vacuum; but then cavitation would occur. Through this the effect of the revolving shaft on the oil layer is diminished, the flow-through being also diminished, so that the pressure distribution changes. The question is: What is the final condition of equilibrium? If air can enter from the rear then the state at which the minimum pressure appears at the pressure p_0, is obviously that which produces the smallest possible flow-through (Fig. 6, curve *b*); then at a still smaller flow-through the minimum pressure would be above atmospheric pressure (dotted curve), which is incompatible with the conditions laid down above. Curve *b* is according to Stieber (1933 *), who assumed that the air enters into the bearing either from the side surfaces or in a tongue shape from behind. Stieber shows that the oil flow is consistent with the condition of continuity only when the pressure gradient is zero. For this, the air-free part of the oil flow must satisfy the condition $dp/dx=0$ at the transition to the state $p=\text{constant}=p_0$.

The question is now: What happens when the oil wedge is enclosed on all sides with oil so that no air can enter? A minimum oil pressure, approximating to a vacuum, was expected. During observation, the measuring pipe was connected to a water-jet pump to prevent air from the pressure gauge entering the bearing. The pump was regulated so as to draw a constant quantity of oil from the bearing, check being kept on the amount by observing the level in a glass tube. With a wide lubricating wedge, under-pressures up to 20 cm. of mercury were observed, whereas with thinner wedges (higher loads) the under-

* Stieber, W., "Das Schwimmlager, Hydrodynamische Theorie des Gleitlagers", VDI-Verlag, Berlin, 1933; see also *Forschung*, 1933, vol. 4, p. 259.

pressure decreased so much as to be hardly noticeable, so that the pressure distribution curve approximated to curve *b* of Fig. 6. The explanation of this behaviour is that the oil in the bearing wedge tends to liberate gas, so that through the reflux of the oil through the narrowest part of the wedge the bubbles are retained near the point of separation. The oil becomes visibly warm at higher loads and thus tends to liberate gas. That the oil when warmed tends to liberate gas has been proved by a special experiment. The conclusion is, therefore, that Stieber's hypothesis is correct for bearings totally enclosed in

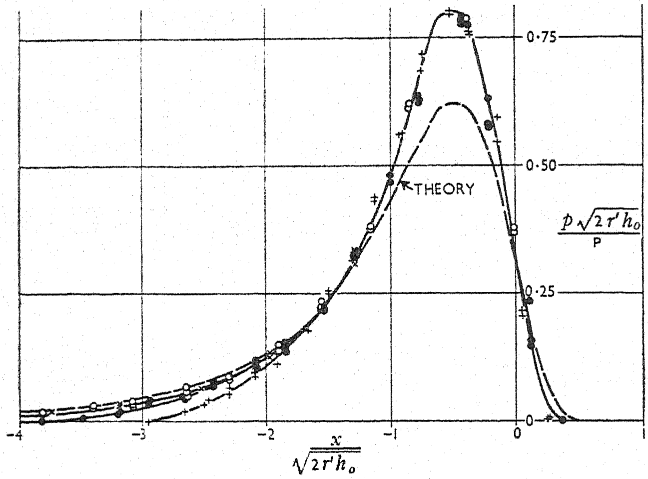

Fig. 7. Non-Dimensional Representation of the Pressure Distribution for $r_2 > r_1$

+ h_o=0·00280 cm. ○ h_o=0·00080 cm.
● h_o=0·00148 cm. × h_o=0·00029 cm.

oil. It is intended to carry out further experiments with "bushes' made of glass.*

Conclusion. These investigations prove that the hydrodynamic theory gives fully correct results, provided that the entry of air into the bearing is prevented, otherwise a layer of air between the oil film on the journal and the bearing brass will considerably reduce the oil film thickness. For bearings where there is a considerable difference between the radii of the journal and the bearing brass, the postulate of Stieber concerning a minimum oil flow is confirmed.

* Since this paper was written, these experiments have been carried out and clearly show the gas production behind the narrowest part of the oil wedge (Sept. 1937).

HEAVY-DUTY BEARINGS OF PHENOLIC PLASTICS

By H. M. Richardson*

During the past eight years there has been a great extension of the use of phenolic-plastic bearings, particularly in the rolling mill industry in the United States. Such bearings have been applied to practically all types of rolling mills except the two-high hot-sheet mill where the rolls are operated at temperatures of 500 deg. F. and the roll necks cannot be water-cooled. They are used on blooming mills, plate mills, bar and billet mills, rod mills, strip mills, skelp mills, tube mills, pipe-welding rolls, three-high sheet mills, cold-sheet mills, structural mills, etc., as well as on such auxiliary apparatus as roll tables, which are normally water-cooled and where the water can be used for lubrication.

The usual practice in the application of the cooling and lubricating water is to provide spray pipes which direct fan-like or conical jets of water against the surface of the roll neck which will provide the utmost cooling effect with a given amount of water. The general system is shown in Fig. 1. Bearings of this sort must have an adequate, dependable supply of cooling and lubricating water. When properly applied and water-lubricated they are very stable, long-wearing, and efficient. If, however, the water supply fails while the rolling mill is in operation, the bearings will soon overheat and char on the surface, and their life will be shortened. But even under such conditions the surface of the roll neck is not scored or otherwise injured.

The amount of water necessary for cooling depends on several factors. First, the amount of heat generated in the bearing. The rate of generation of heat at any instant is $f \times p \times s/778$ B.Th.U. per square inch of bearing area per minute, where f is the coefficient of friction, p the pressure in pounds per square inch, and s the rubbing speed in feet per minute. In other words, the heat generated increases directly as the coefficient of friction, the speed, and the pressure. Second, the duty cycle—whether the pressure and speed are intermittent, with intervening periods of light loads. Third, the amount of heat transmitted to the roll neck from the roll body and the coupling. Fourth, the efficiency with which the water carries away the heat from the neck surface. Cooling and lubricating water is applied by providing a spray pipe on each side of the bearing. These pipes are designed to bring as much of the cooling water as possible into close contact

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with the roll neck. The spray should be directed to all parts of the open surface of the neck. One way of accomplishing this is to make the spray pipe with diagonal slots which allow a fan-shaped spray to be directed against the neck. Another method is to provide two or three rows of holes in the spray pipe to distribute the water more evenly on the neck surface.

Suitable strainers should be provided in the water line to lessen the chance of stoppage of the spray orifice. All new piping should be carefully burred and all burrs which might catch particles of dirt or lint

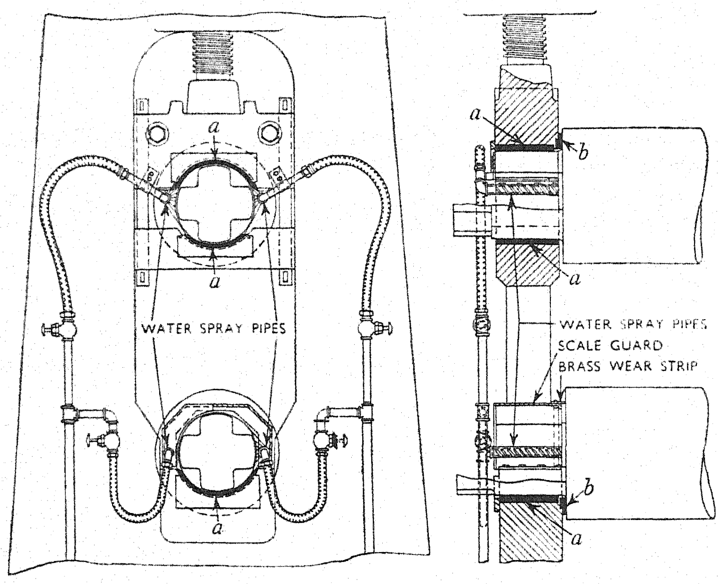


Fig. 1. Outline of Mounting for Phenolic Plastic Bearings with Water Lubrication

a Liner. *b* Thrust collar.

should be removed from the inside of the spray pipe. The piping should be well supported so that the spray pipes cannot be accidentally dislodged from their proper position; but the pipes should be easily removable to save time when the rolls are changed.

When the mill stand is assembled or the rolls are changed, a coating of heavy lubricating oil should be put on the roll neck to provide a lubricating film when the mill is started. Further, when the mill is stopped for several hours the roll necks should be coated with heavy oil to protect them from rusting and to lower the starting friction.

One way to apply this protection at the time the mill is shut down is to stop the mill, turn off the water, apply some heavy lubricating oil to each roll neck from a hand-operated grease gun, and then to turn the rolls three or four revolutions to spread the grease film over the neck and the bearing surface. This is usually the only time when it is necessary to use grease on the roll necks. Apart from rust prevention, this greasing after stopping also seems to stabilize the bearing friction for a time after the mill has been started with the regular water lubrication, until the bearings are run-in to a full contact with the roll neck.

For best results the purity of the water requires consideration. The water supply should be reasonably free from suspended sand or scale. A large amount of sand or scale will roughen the neck surface and cause more rapid wear of the bearing. In places where there are likely to be large amounts of black scale from the steel, as on the bottom rolls of blooming, slabbing, and plate mills, an effective scale guard should be installed to keep out most of the scale and allow effective water lubrication. One type of scale guard is shown in Fig. 1. An extra scale guard should be kept on hand so that, in case of accidental damage to the one installed, it will not be necessary to run the mill without scale protection for the bottom roll neck. Acids or salt in the cooling water will often corrode the surface of the roll neck and prevent the maintenance of a smooth journal surface. This results in shorter bearing life and prevents the usual polished surface from being formed on the roll neck. The effect of corrosive agents in the water is more pronounced as the water temperature rises above approximately 90 deg. F.

The usual practice in the United States in the application of phenolic laminated roll neck bearings is to take the existing bearing blocks or chucks and machine them out to accommodate a radial bearing lining which is a cylindrical segment, and a separate thrust collar which is machined from a flat slab of the material. Several examples of this type of construction are shown in Figs. 2–6. It is desirable, whatever system of adaptation is used, to provide solid backing for the bearing lining and thrust collar, and suitable anchorage so that they are not likely to be torn loose in case of high frictional drag caused by loss of lubricating water.

If new bearing chucks are made, the type shown in Fig. 2 is suggested. The existing chucks often can be machined out and fitted with keeper keys (Fig. 3), or with bolted-down keeper plates (Fig. 2). Some applications, such as cold-sheet mills, require steel adapters, fitted with the bearing lining, to take the place of heavy slab-like bronze bearings. The methods shown in Figs. 4 and 5 are satisfactory

in this case. The old bronze bearings are sometimes machined out and faced with phenolic laminated material (Fig. 6), dowels of the same type of material being used to anchor the linings in place.

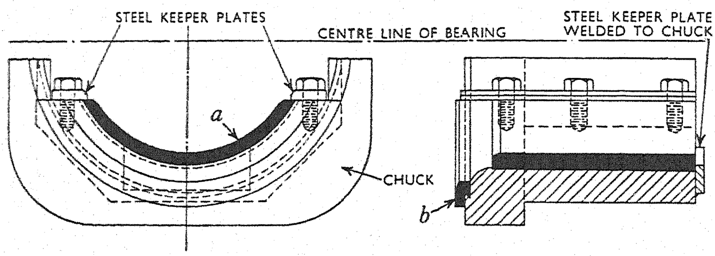


Fig. 2. Type of Phenolic Laminated Roll Neck Bearing

a Liner. *b* Thrust collar.

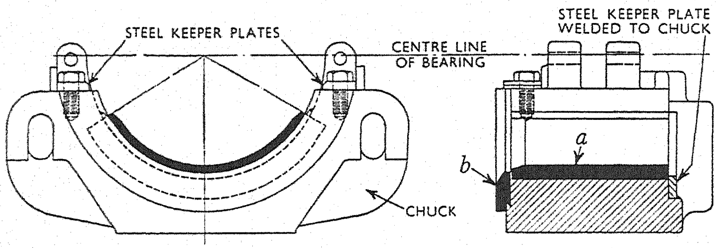


Fig. 3. Type of Phenolic Laminated Roll Neck Bearing

a Liner. *b* Thrust collar.

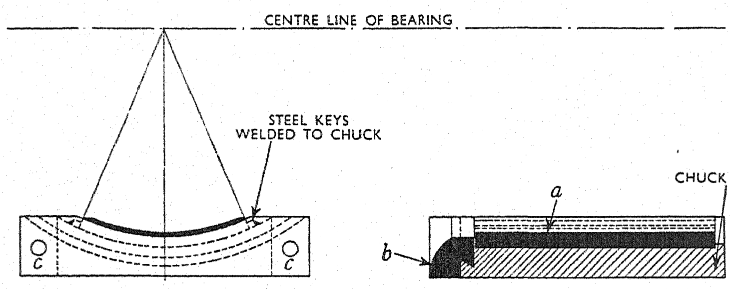


Fig. 4. Type of Phenolic Laminated Roll Neck Bearing

a Liner. *b* Thrust collar. *c* Dowels.

In some mills, particularly in those which are not subjected to extremely high pressures or severe pounding, and in which there is a substantial number of identical bearings (36 or more, for example), it

may be economical to invest in a mould for making the radial lining and thrust collar in a single piece of the same shape as the bronze bearings to be replaced. These bearings are usually somewhat more expensive than the simple linings and thrust collars (Figs. 2–6), but the cost of making new chucks or machining the old ones, if there are

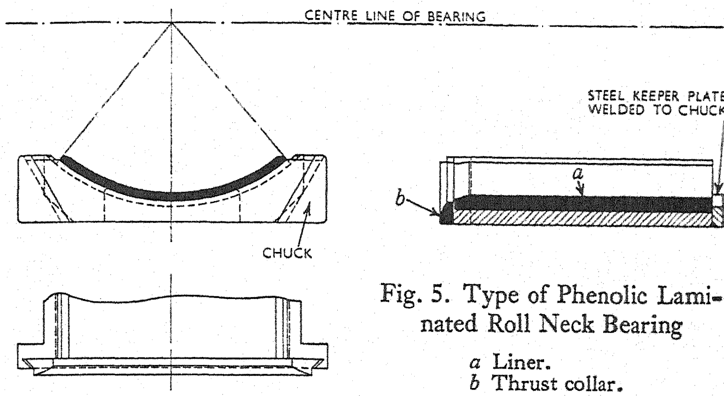


Fig. 5. Type of Phenolic Laminated Roll Neck Bearing

a Liner.
b Thrust collar.

many of them, may offset this difference plus the cost of the mould. Before deciding to use moulded bearings, it is advisable to examine carefully the chucks which are to hold the bearings, in order that the new bearings may have good support, and to make sure that they are not badly worn or distorted. If the chucks are badly worn, it may be advisable to make new ones, like that shown in Fig. 2. This eliminates

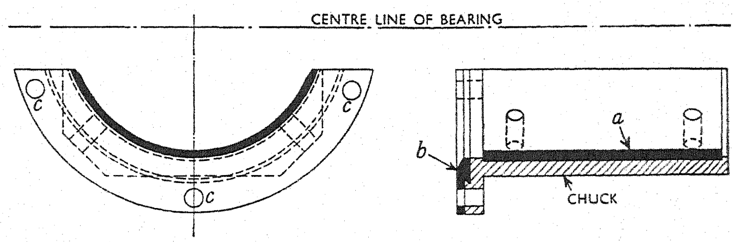


Fig. 6. Bronze Bearing Faced with Phenolic Laminated Material

a Liner. *b* Thrust collar. *c* Dowels.

the cost of a special mould, and takes advantage of the lower replacement cost of the simpler linings and thrust collars.

In some mills, such as plate mills with side bearings, it is economical to machine the bearings from slabs of phenolic laminated material, except for top or bottom bearings where heavy direct pressure is

carried and where it is desirable to use a thin lining to keep compressive deflexion to the minimum. In such cases the arrangement in Figs. 4 or 5 is preferred.

The proper thickness of bearing linings for a good installation is usually determined by the amount of material which must be machined out of the old metal bearing or chuck to give the phenolic laminated material a good solid backing. This is the best procedure to follow unless the resulting lining is thick enough to add an objectionable amount of "spring" or elasticity to the mill. The amount of this spring, or compressive flexibility, in the phenolic laminated material is roughly 0·002 inch per 1,000 lb. per sq. in. bearing pressure per inch of lining thickness. This is usually not an important problem, except for hot-strip mills, where temperature variations in the strip result in wide variations of rolling pressure. The strip must be held to precise dimensions and is usually rolled to a thickness near the maximum. If there is too much spring in the mill, the cooler parts of the strip may be too thick. For mills of this type, the maximum thickness of the phenolic laminated bearing lining is about $\frac{5}{8}$ inch. The usual range of lining thickness for other mills is from $\frac{1}{2}$ inch to $1\frac{1}{2}$ inches.

Another factor in design is the arc of contact of the bearing. As a result of the very low friction characteristics of phenolic laminated bearings, the arc of contact required to maintain proper alignment may be much less than with metal bearings. This reduced arc provides more economical operation, since it is desirable that as much as possible of the roll neck should be open and in contact with the cooling water. For roll necks without side bearings, this arc of contact ranges from a minimum of 90 deg. to a maximum of about 135 deg. Single stands (not reversing) should use 90–120 deg., while stands in continuous mills or reversing mills should use bearings of from 120–135 deg. Large mills with side bearings should use top and bottom bearings of 60–90 deg. arc.

Thrust collars should be securely anchored to the face of the chuck or adapter. They are usually bored out to a radius larger than that of the roll neck to allow a channel for the flow of cooling water to the face of the collar. Two or three radial grooves across the face of the thrust are desirable on large collars. In mills rolling shapes or rounds, where it is the usual practice to hold a very tight mill, there is a tendency for thrust collars to wear out rapidly until the roller learns the "feel" of non-metallic bearings. The reason is that the resilient, non-metallic thrust collars will compress about twenty times as much under a tight end-adjustment as a corresponding bronze or Babbitt collar. This means that much greater wear occurs in the non-metallic thrust collar than in the metal thrust collar before it clears itself enough to allow

the lubricant to enter. This condition can be helped considerably by designing the thrust collars to give as large an arc of contact as possible. This may be carried to the limit by making a chuck that provides a full-ring thrust collar, which has been done in some cases with considerable success. The best results cannot be obtained, however, until the roller operating the mill learns the "feel" of the bearings, so that he can adjust the mill tightly enough to hold the rolled shape to an accurate cross-section, but not so tightly that the thrust collar is worn out rapidly. Experience has demonstrated that the first collars are often short-lived, but that each succeeding set lasts longer as the roller learns to adjust his mill properly, until the life of the collars finally is comparable with that of the radial linings.

Another factor affecting the life of the thrust collars is the amount of backlash in the mill. It is true that a mill with properly adjusted phenolic laminated bearings runs very freely, so that a rattle often develops in the couplings. This rattle is sometimes taken out by tightening the thrust collars, which shortens their life. If the rattle cannot be removed by improving the roll alignment or by improving the fit of the couplings, it may be necessary to put a brake on the roll neck. This brake may consist of another bearing held under spring pressure against the upper side of the bottom roll neck, or, as in some mills, mounted outside the housing and coupled to the outer ends of the rolls. If the need for this brake can be avoided by improving the couplings, this is the better method, because any such friction drag adds to the cost of operation since it uses up some of the power which is saved by the normally low bearing friction.

Although the principal field of use for phenolic laminated bearings has been in the rolling mill industry, there are other fields of application in which they can be used economically. In the manufacture of paper, water-lubricated bearings can be used in many of the wet processes. In the manufacture of cane sugar, some economies can be obtained by using phenolic laminated bearings on the crushing rolls. Here the speed of operation is very slow, giving a sliding velocity of about 12 ft. per min. In this case grease lubrication should be used, because the velocity is not high enough to establish a lubricating film of water.

In general, all the present successful applications of phenolic laminated bearings fall into one or another of the following four classifications :—

(1) Water-lubricated bearings which must be provided with plenty of clean water, adequate provision for circulation of the water, means of preventing corrosion during stoppage, and journal surfaces of reasonable hardness.

(2) Grease-lubricated bearings which are used intermittently and are not often lubricated (the bearings of a crane, for example). Grease must be used sparingly in order to avoid drip. Service is intermittent; hence, heat due to friction can be carried off easily through the shaft. Scanty grease lubrication does no particular harm, since the bearings will not seize and injure the shaft.

(3) Bearings subjected to heavy impact loads in intermittent service. These bearings are grease- or oil-lubricated and have an overall load-speed duty cycle which allows the heat of friction to be carried off by the journal. The advantage of phenolic plastic bearings in such applications is that they neither "pound out" like Babbitt bearings, nor crystallize and cause scoring like bronze bearings.

(4) Oil-lubricated bearings with circulating-oil lubrication in which the heat of friction can be carried away from the bearing by the oil and by the journal. Here the phenolic plastic bearing has no particular advantage over good Babbitt bearings, except where unusually heavy loads or impacts are encountered.

In the course of their application over the past few years specialized grades of phenolic laminated bearing materials have been developed. The desirable characteristics of these bearings are :—

(1) Resistance to heavy impact loads.

(2) Uniformity of structure which allows the bearing to run-in to a very smooth bearing surface.

(3) A very thorough polymerization of the bonding resins so that local heating on the surface of the bearing under heavy pressures will not cause the softening of the material, with consequent gumminess which greatly increases the friction. Much work has been done to bring about heat stability of these bearing materials by the addition of certain accelerating agents to the bonding resin, so that the bearings can be used at extremely high pressure and with a very low coefficient of friction.

Economies Resulting from the Use of Phenolic Plastic Bearings. The many economies which have resulted from the use of phenolic plastic bearings have been responsible for their widespread use, particularly in the rolling mill industry. Paramount among these economies is the saving in power due to the very low coefficient of friction of such bearings with water lubrication. Savings of 15–40 per cent in the

power required to drive rolling mills are common; in some cases the savings have been as high as 60 per cent. Not only do these bearings give substantial savings in power, but they have a much longer life than the bronze or Babbitt bearings previously used. When properly applied and lubricated these phenolic laminated bearings outlast the metallic bearings by from 3 to 10 or 15 times.

Other minor savings are brought about by the elimination of scoring and overheating of the roll necks or other journals which are operated on these bearings. These three principal economies are supplemented

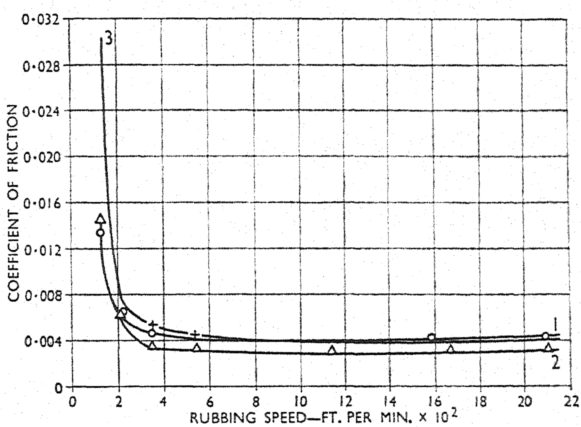


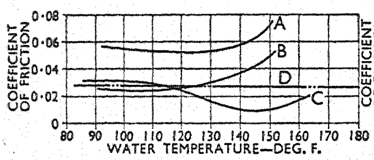
Fig. 7. Friction Curves for Phenolic Plastic Bearings with Water Lubrication

Bearing pressure, lb. per sq. in.:—
1 ○ 3,000.
2 △ 3,500.
3 + 4,000.

by savings due to the reduced cost of maintenance of equipment and reduced cost of lubrication.

Technical Reasons for the Success of Phenolic Plastic Bearings. After examining many tests of phenolic laminated bearings there appear to be two fundamental reasons for their success. The friction curves of Figs. 7, 8, and 9 show that very high bearing pressures can be carried at coefficients of friction so low that film lubrication must take place. When the characteristics of phenolic laminated materials are examined, to discover an explanation for the stability of this low friction, it is found that the material has a very low modulus of elasticity (approximately 500,000 lb. per sq. in. per in. deflexion). According to the

usual theory of oil film lubrication where rigid bodies are assumed, the pressure within the oil film depends upon the viscosity of the fluid, the thickness of the film, the relative velocity of the bearing and journal surfaces, and the angle between the two surfaces. With rigid bodies there is a considerable concentration of the bearing pressure at the point where the lubricating film is of least thickness. These conditions are changed where the bearing is made of material which is many times more elastic than the usual metal bearing. In this case, as the pressure within the lubricating film increases, or tends to concentrate at the point of minimum thickness, the pressure is sufficiently high to cause



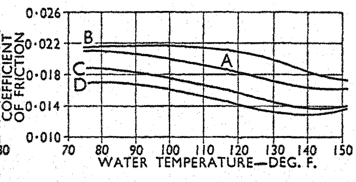


Fig. 8 Fig. 9

Friction Curves for a Phenolic Plastic Bearing

Steel shaft, case-hardened; Shore hardness No., 45–55.

Curve	Lubrication	Water, gal. per min. per sq. in.	Bearing pressure, lb. per sq. in.	Rubbing speed, ft. per min.
Fig. 8:—				
A . .	Water	3·5	4,000	91·5
B . .	„	3·5	6,000	150
C . .	„	3·5	6,000	250
D . .	Grease*	0·65	6,000	80
Fig. 9:—				
A . .	Grease*	0·61	9,000	80
B . .	„ *	0·61	9,000	150
C . .	„ *	0·61	8,000	80
D . .	„ *	0·61	8,000	150

* Water cooling.

appreciable deflexion or compression of the bearing lining, distorting it in such a way that the oil film does not decrease in thickness as much as it would with a rigid bearing under the same total loading. The net result is a more even distribution of pressure throughout the lubricating film and less concentration of pressure. The tendency is toward stability.

The non-metallic chemical structure of phenolic-plastic bearings is such that when brought into close contact with the metal they do not tend to adhere. Even though phenolic laminated bearings are operated

without lubrication the heating which occurs does not result in scoring or "galling" of the surface of the journal. The greatest damage which can occur is the charring of the surface of the non-metallic bearing.

Limitations of the Use of Phenolic Plastic Bearings. Phenolic plastic bearings are not universal in their application as they have certain definite limitations. First, they have a very low heat conductivity (approximately 2 B.Th.U. per sq. ft. per in. thickness per hr. per deg. F.). The heat conductivity of bronze is about 400 times as great. In designing phenolic plastic bearings provision must therefore be made for carrying away the heat of friction either in the lubricant or through the journal. The relative heat conductivity of the bearing material itself is so low that it can be considered as a heat insulator rather than as a heat conductor.

The other limitation is one of temperature. Phenolic laminated bearings with cotton fabric as the filler should not be used at temperatures in excess of 300 deg. F., because at higher temperatures the material begins to char. If asbestos fabric is substituted for the cotton fabric in such bearings and precaution is taken to cure the bonding resin thoroughly, this temperature limit can be raised to about 400 deg. F.

As time goes on, probably more and more applications will be made of phenolic-plastic bearings, not only in applications which involve high pressures, but also in other applications where lighter loads are encountered. In any event these two limitations will still apply.

FABRIC BEARINGS FOR ROLLING MILLS

By H. Rochester *

The use of fabric bearings for rolling mill work is now well established and the demand continues to grow as their advantages are more fully realized. The three main advantages are:—

(*a*) Reduction in friction with great saving in the driving power costs.

(*b*) The rolls can be held to gauge setting, giving a reduction in cost out work.

(*c*) Reduction in maintenance, and longer runs without shut-downs.

The incidental advantages are: cleaner operation, grease consumption minimized, the roll necks are not damaged, and both neck and roll are worked at a lower temperature and are therefore not so subject to heat cracks, less attendance is wanted as only an ample supply of cooling water is required, continuous adjustment is unnecessary, more accurate work results, loading can be increased and heavier reductions made, so that the number of passes required is reduced and the output can be increased.

The chief requirement is a continuous ample supply of cooling water, applied so that the roll necks are always in contact with the incoming cold water. The best method is to spray the neck with several jets, so that the bearing receives cold water for lubrication and the cylinder of hot water adhering to the neck is broken and carried away without re-entering the bearing. These precautions are necessary due to the poor heat conductivity of the fabric. The scheme is shown in Fig. 1.

The use of fabric bearings is in no way restricted to rolling mill work, and can be used in a variety of other applications, always with the provision of the necessary means of cooling.

Characteristics.

Specific gravity, 1·25–1·4.

For general calculations take 1 lb.=20 cu. in.

Brinell hardness No., 30 to 42.

Water absorption, 7-day test, 1 to 3 per cent.

Compressive strength at the end of layers, 30,000–35,000 lb. per sq. in.

Compressive strength on flat of layers, 45,000–50,000 lb. per sq. in.

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* Metropolitan-Vickers Electrical Company, Ltd., Manchester.

Compressive strength, moulded clippings, 25,000–30,000 lb. per sq. in.

Compression per inch thickness per 1,000 lb. per sq. in., 0·0018–0·002 inch.

Tensile strength along grain, 8,000–15,000 lb. per sq. in.

Tensile strength across grain, 6,000–11,000 lb. per sq. in.

Tensile strength, clipping material, 2,800–3,200 lb. per sq. in.

Modulus of elasticity (sheet), 750,000 lb.

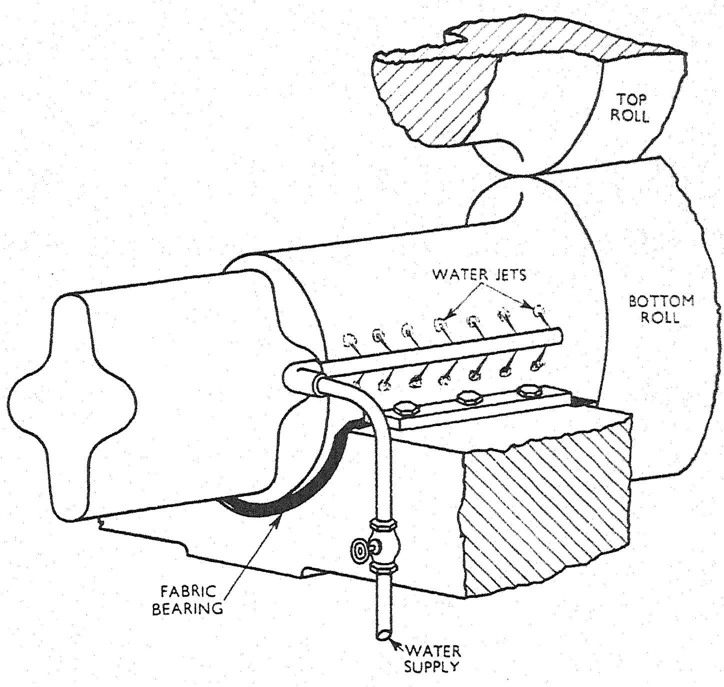

Fig. 1. Method of Applying Cooling Water

Modulus of elasticity (moulded), 280,000 lb.

Splitting resistance (sheet), 1,450 lb.

Splitting resistance (moulded), 1,200 lb.

Moulding temperature, 150–160 deg. C.

Maximum allowable running temperature for short periods, 250 deg. C.

Recommended maximum working temperature, 60–80 deg. C.

Loading. It can be taken in general that if the bronze or Babbitt bearings are being replaced by fabric bearings they will be easily

capable of carrying the load, and as they are very much more resilient, no hardening and cracking, as in the case of bronze, or on the other hand, creeping or flowing as in Babbitt bearings, under the hammer action of the work entering the rolls, will occur. Further, the fabric acts as a polishing pad and helps to keep the roll necks in good condition. It is suggested that loading be divided out as follows:—

Light: All pressures up to 500 lb. per sq. in.
Medium: All pressures from 500 lb. to 1,500 lb. per sq. in.
Heavy: All pressures from 1,500 lb. to 4,000 lb. per sq. in.
Extra heavy: All pressures over 4,000 lb. per sq. in.

Lubrication and Cooling. As the fabric is a poor heat conductor, approximately 500 times less than that of metal, it is necessary to ensure that sufficient lubricant is in intimate contact with the metal neck to cool the bearing and to carry away the heat generated, as well as to serve for lubrication. This requirement does not preclude any form of liquid being used, but as oil and expensive liquids would have to be collected, cleaned, cooled, and pumped back again through the bearing, it is noteworthy that water forms a completely satisfactory medium over a wide range of loads and rubbing speeds. A water film in a bearing can be broken by slow speed, or by heavy pressure, and this limits the use of water approximately as follows:—

With light loads the range covered is from 100 to 4,000 ft. per min. rubbing speed; with medium loads the scale is shortened to 150 to 3,000 ft. per min. rubbing speed, and with heavy loads from 250 to 2,500 ft. per min. rubbing speed, but the scale can be extended at both ends by using grease to help to lubricate and hold a water film. Extra heavy loads need grease with water, or sud-oil with water all the time.

The sud-oil can be of any standard make of emulsifying oil, similar to those used on capstan lathes as a tool-cooling fluid, and can be used at the rate of from 5 to 15 per cent sud-oil in water, in accordance with the load. Grease in every case is reduced to a very small quantity compared with that required for metal bearings, and where it is only required to prevent rusting during the standing periods, a cheap oil or grease can be used; but it is better to avoid anything of a tarry nature which would stick to the shaft or fabric and carry grit, or prevent free admission of the cooling water.

Friction and Power Saving. The coefficient of friction is fairly low compared with that of metal bearings, but depends on the coarseness of the fabric used in manufacture. The finer cloths give a low coefficient, increasing almost directly in proportion to the coarseness of the cloth, and the load carried increases with the coarseness of the cloth as a

general rule, so that in the selection of a bearing, to obtain the maximum saving, it is necessary to grade the material in accordance with the load to be carried. For very light loads, fabric does not compare well with Babbitt metal with oil lubrication.

Fig. 3 gives the results of tests carried out on a normal standard oil ring bearing, while Fig. 4 gives the friction at various loads and rubbing speeds with water lubrication only, for a medium fabric.

The quantity of cooling water required depends on the following factors:—

(1) The amount of heat generated in the bearing.

(2) The load cycle, where loading speed is steady or variable, and the length of runs and time of rests.

(3) Heat transmitted to neck from the work passing through the rolls, and the efficiency of application of the cooling water.

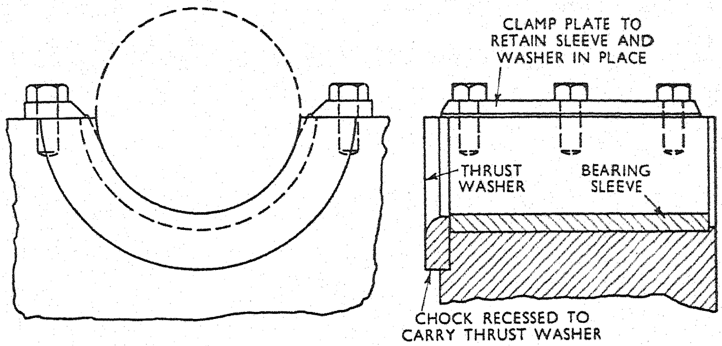

Fig. 2. Standard Mounting for Fabric Bearing

A rough rule for the quantity of water required in gallons per hour per bearing may be taken as equal to: $(D \times L \times \mu \times \text{r.p.m.})/16{,}000$, where D=the diameter of the neck in inches, L=the total load on the bearing in pounds, and μ=the coefficient of friction, which for normal calculations may be taken as 0·008.

The saving in power required to drive a mill owing to the use of fabric bearings is due to (*a*) the lower coefficient of friction, and (*b*) the fact that the necks can be made and maintained in a better condition. If the neck and thrust faces are ground and polished before fitting the fabrics, this condition will remain permanently, as the bearings act as a polishing pad and keep the bearing surfaces in good condition. In practice, savings up to 50 per cent in power have been attained and maintained in several rolling mills, and it is seldom that the saving is under 30 per cent.

If desired, some of the saving can be used by increasing the output of the mill, as the bearings will carry heavier loads, so that greater reductions per pass can be made, thus reducing the time required for a given output. The use of fabric bearings has also made it possible to roll wider planished strip. Fine sheet which demands preloading on the bearings can also be rolled to closer gauge, with considerably less adjustment than for metal bearings.

Applications. Fabric bearings can always be used to replace metal bearings, the general requirement being a plentiful supply of water. The best form in which to use the fabric is to mould the bearing sleeve directly to shape, and to cut the thrust washers from sheet, so that the flat of the material is presented for wear. Fig. 2 shows an approved method of application.

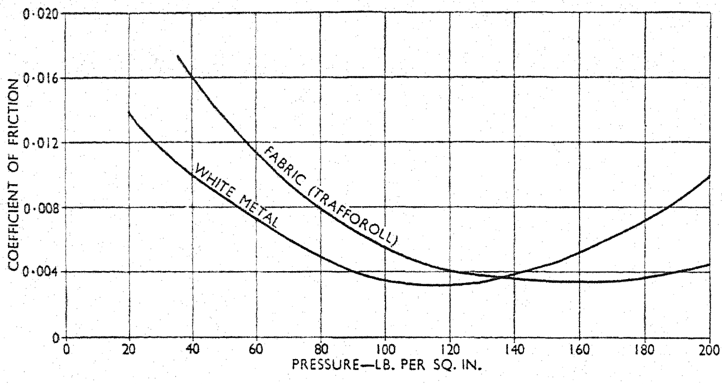

Fig. 3. Friction Tests carried out with a Standard Oil Ring Bearing (5 inches Diameter) with Oil Lubrication

The radial thickness of the liner may vary between $\frac{3}{8}$ to $1\frac{1}{4}$ inches, increasing with the neck diameter. The thickness of the thrust washer should be at least as great as the radius in the roll neck. The angle embraced by the sleeve should be approximately as follows:—

Single stands without side bearings .	90–120 deg. arc of contact.
Continuous and reversing mills without side bearings	120–140 deg. arc of contact.
Large mills with side bearings . .	80–100 deg. arc of contact.

The fabric should be held in position by clamping plates, and well bedded and supported by the chocks. Alignment should have due attention, and the whole set-up should be carefully done, instead of the not unusual practice of allowing rough fitting and "self grinding-in".

These points must be observed if full advantage is to be obtained, with long life from the fabric material. The expected life of a fabric bearing is not less than four to five times that of a metal bearing, while ten to twelve times is common, with rare cases of over twenty times.

Starting and Running. Under no circumstance must a bearing be run without the water being first turned on and running freely. If the bearing is likely to be dry or heavily loaded, a little grease will help in starting; and when shutting down for any length of time, if during the last few revolutions of the mill a little grease is rubbed in, this will prevent rusting during the standing period, and the mill will be ready

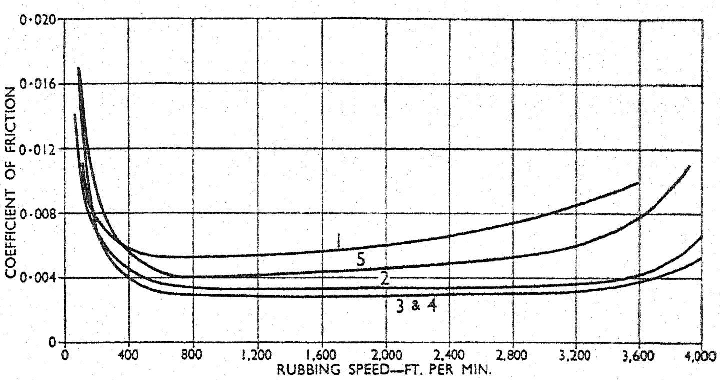

Fig. 4. Friction at Various Loads and Speeds with Water Lubrication Only

Pressures :—

Curve 1	600 lb. per sq. in.	
,, 2	1,000 ,,	,,
,, 3	2,100 ,,	,,
,, 4	2,900 ,,	,,
,, 5	4,000 ,,	,,

for starting again when required; of course, this is not necessary for short stops.

When starting up a new set of bearings, always grease the roll neck with a heavy grease or cylinder oil, and barr the rolls around a few turns to spread the grease, then turn on the water and when this is freely running give the mill a short run on light load, to allow the bearings to settle and bed in.

The bearing surface will gradually burnish, and when this condition has been reached the bearing is in its best condition, and care should be taken to maintain it so.

THE LUBRICATION OF LAMINATED SYNTHETIC RESINOID BEARING MATERIALS

By Herbert W. Rowell*

Synthetic resinoid bearing material is made by impregnating cloth or paper with synthetic phenolic resinoids and then compressing a large number of layers under hydraulic pressure in heated steel moulds to the shape required, or to obtain a block which is subsequently machined to form. The chemical change in the resinoid during heating converts the laminated plastic mass into an infusible, tough, and hard-wearing composite material.

This material has been particularly successful when used on large rolling mills as water-lubricated, roll-neck bearing blocks of the usual form, or as liners to the metal blocks. Compared with metal bearings, there is a very considerable saving of power, less wear, and less stoppage for replacement, with consequent better operation of the mill. This success is due to the peculiar properties of the material which enable it to hold a much heavier film of water lubricant than a metal can at a similar speed and high bearing pressure.

In Germany it is claimed that over 80 per cent of rolling mill bearings are made from this material and a considerable saving of power is reported. In America, as a result of several years' experience, a similar high saving of power per ton of metal rolled is reported. The approach of British engineers to the material has been careful and conservative, but many such bearings have been fitted, with considerable profit, during the past two or three years. There are many other profitable applications of water-lubricated bearings.

At first sight it seems absurd to use an efficient brake shoe lining as a bearing surface. When dry, the material has a coefficient of friction equal to that of good automobile brake linings. When lubricated with water, however, the coefficient of friction is much lower than that of metals lubricated with water, if test conditions are the same. No laboratory figures are offered because they may be misleading. The determination of the coefficient of friction involves many possibilities of experimental error and the figures obtained by different operators using different methods do not yet compare well between themselves or with practical experience.

In rolling-mill practice, with water lubrication, the saving in power with these synthetic bearings is reported at from 15 to 80 per cent. The higher claim probably includes improved fitting or mechanical

* Ellison Insulations, Ltd. (Tufnol).

conditions, but a safe assumption, comparing this synthetic material with bronze, would be 20 to 30 per cent. Obviously, this saving in power can be due only to more efficient lubrication reducing the coefficient of friction under working conditions. It is axiomatic that the face of the bearing cannot wear if the lubricating film remains effective. It is also well known that if the journal is accurately fitted to the curvature of the shaft, the film of lubricant is of even thickness, can be maintained at maximum thickness and lowest coefficient of friction, and is not so easily broken as in a badly fitting bearing.

The reason for the superiority of synthetic resinoid bearing material over metal lies in the fact that it is a composite of organic fibres and hard resinoid, which will absorb water but will not absorb oil or grease. The fibres absorb the water while the resinoid does not, and the absorption at a machined surface which exposes cut fibres is of the order of 0·03 to 0·05 oz. of water per sq. ft. of wet surface in 24 hours. Immersion during several months is required to wet the fibres lying ½ inch below the surface.

It has not yet been demonstrated that a bearing of this material has a consistently lower coefficient of friction than metal when both are lubricated with oil or grease. Its considerable superiority when both are lubricated with water is amply demonstrated in practice and it is also superior, if water-lubricated, when compared with metal lubricated with oil.

The outer molecular layers of a lubricating film attach themselves with very considerable force to a polished metal face and do not form part of the active lubricant. The centre portion of the film forms the mobile or viscous mass on which the bearing rides and it follows that the thicker the viscous part of the film, the less will be the friction and heat produced.

The working surface of a laminated material is unlike a polished metal surface in that it consists largely of a forest of microscopic cellulose fibres impregnated with water. A force much greater than that produced in a heavy bearing would be required to squeeze the water out of these colloidal fibres. The "trees" in this forest are not more than 1/1,000 or 2/1,000 inch high and about 1/10,000 inch thick. There are patches of bare resinoid between patches of fibre; and under heavy loads the vertical fibres are probably bent over and laid flat, so that a tough jelly-like mat of wet, elastic fibre forms the face of one of these bearings in operation. Water is also held between the fibres by the strong forces of surface tension or capillary attraction, and the films thus formed assist in retaining the mobile lubricating film of water above them.

This picture gives a clue to the mechanism by which a more ser-

viceable film of water is maintained at high bearing loads between the steel shaft and the synthetic bearing material. The peak pressures are sufficient to break the thin film of lubricating water which can be held by molecular adhesion on metal surfaces and the frequent contact of metal to metal rapidly wears the softer bearing face. Study of the fibrous face still continues in order to discover whether the saturated cellulose also acts as a lubricant at the same time as it maintains the more effective film of water.

These synthetic organic materials naturally begin to burn and wear more rapidly at temperatures only about 50 deg. F. over that of boiling water, and water lubrication is therefore not only most effective but also a necessary and cheap cooling medium at high bearing pressures. On heavy loads with oil lubricant, the removal of heat can be effected by pump lubrication, but on light loads, ordinary methods of lubrication and heat dissipation are generally sufficient, and the bearing surface becomes well burnished.

Graphite lubrication of a dry bearing on light load can be quite effective because bearings run-in with a liberal supply of graphite retain a good coat on the fibrous surface. If it is possible to incorporate sufficient graphite in such a composite material without reducing its mechanical strength, it obviously cannot maintain any more effective supply of graphite unless the bearing material wears and releases it. Materials of this type which have been tried, have still to prove their superiority over the graphitized surface of a plain material, but graphite lubrication is attracting considerable attention.

The discussion of other properties and uses of this material as a bearing surface is dealt with by other papers in this series.

LUBRICATION OF JOURNAL BEARINGS WITH WATER-BASE LUBRICANT

By F. Samuelson, M.I.Mech.E.*

Disastrous fires have been reported in electrical power stations, due to the burning of the lubricating oil used in the turbo-generator system. In a large set some 2,000 gallons of oil would be in circulation, and the burning of this quantity of oil would be a very serious matter to the plant.

With a view to overcoming the liability to fire, the British Thomson-Houston Company, Ltd., have been experimenting for some time, using as a lubricant a water-soluble oil. Observation of the action of this lubricant on heavy cutting tools suggested its application to the lubrication of steam turbine bearings. Steam temperatures have increased very rapidly lately and oil coming in contact with the exposed high-temperature parts of the turbine would immediately ignite.

The first experiments were made upon a 25 kW., 3,000 r.p.m. shop turbine set, which has been running quite satisfactorily on a water-base lubricant for about 30 months. The oil pump clearances and governor oilways had to be reduced and made finer than is normal, otherwise no modifications were made to the general design or material of bearings or other parts. The governor operates quite satisfactorily at a pressure of 50 lb. per sq. in.

To be satisfactory a lubricant of this class must not cause any chemical action directly or through electrolytic action and must be non-inflammable.

Various soluble oils were tried, until finally an oil having the following approximate composition was decided upon:—

Mineral lubricating oil .	50	per cent (highly refined, first quality)
Moisture	13½	,,
Antiseptic	2½	,,
Dry soap	28	,,
Free fatty acids . .	6	,,

One part of this oil was diluted with 50 parts of water. The viscosity at 60 deg. F. is approximately 29 sec. Redwood. Ordinary tap water was used for dilution.

In the small shop turbine the bearing loads are small and the journal speeds low, so a further test was made in the standard bearing testing machine, Fig. 1, using a journal 6 inches in diameter and 9 inches long, and various loads and speeds such as are usual in standard turbine practice. The test equipment consists of a journal shaft supported by

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* The British Thomson-Houston Company, Ltd.

the bearing under test. Variable loads can be imposed on the shaft by means of a lever and knife-edge. Fig. 2 shows the journal bearing,

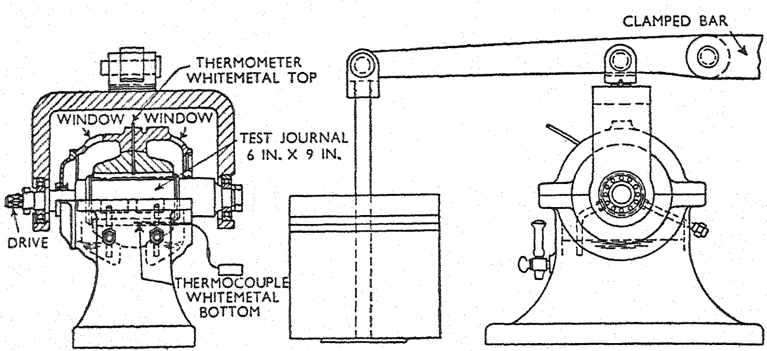


Fig. 1. Bearing Test Apparatus

together with the lubricating passages. The arc of contact for the bottom half was approximately 100 deg. The bearing was bored to

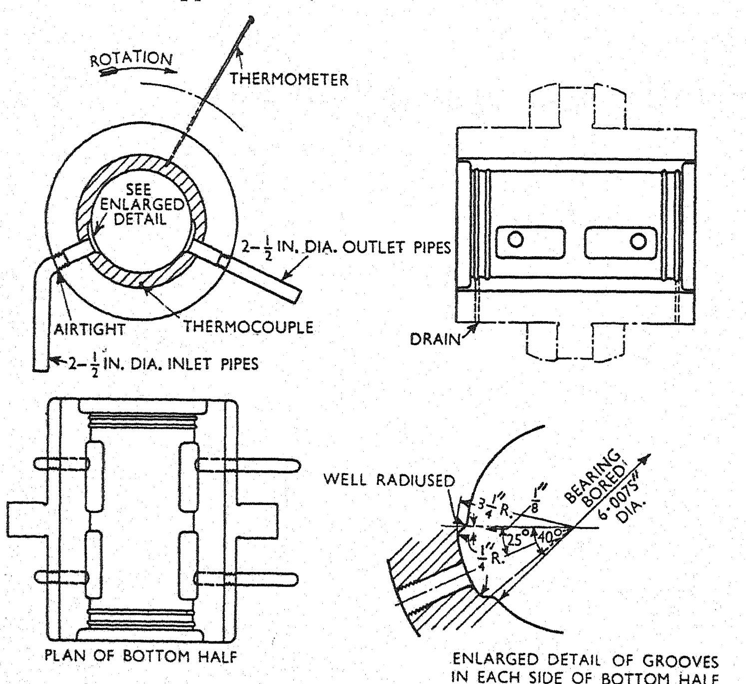


Fig. 2. Experimental Self-lubricating Bearing

give an eccentric clearance of 0·0015 inch per inch diameter. The lubricant enters the bearing by means of two ½-inch pipes, 7 inches apart, and flows into two short lengths of troughs ¼ inch deep. Similar troughs and piping allowed the spent lubricant to be discharged from the bearing ways on the opposite side of the bearing to the inlet.

Fig. 3 illustrates the lubricating arrangement. It should be particularly observed that the shaft rotating in the bearing itself pumps the oil without any external aid from an auxiliary pump. The lift to the centre of the journal was 6–12 inches. Figs. 4–6 illustrate the behaviour of the bearing under various load and speed conditions. Fig. 4 also includes results from tests on a 6 × 9-inch bearing lubricated

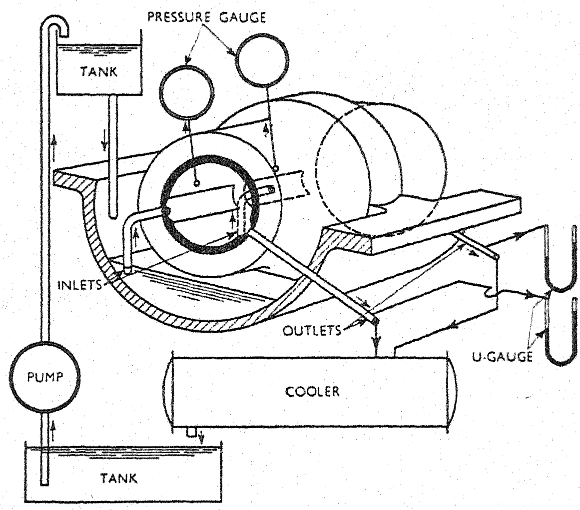


Fig. 3. Diagram of Connexions in the Bearing Test

in the normal way by circulating 30 lb. per min. of oil. It will be observed that at 6,000 r.p.m. the whitemetal at the bottom of the bearing was at 130 deg. F. with the special water lubricant, whilst for oil the metal was at 182 deg. F., i.e. 40 per cent higher.

Fig. 5*a* shows the losses as registered by the driving motor, and as obtained by thermal calculation for a constant load at increasing speeds from 3,000 up to 6,000 r.p.m. and corresponds with the temperature rises of Fig. 6. Fig. 5*a* also shows for the same size bearing the corresponding oil losses when fed at 30 lb. per min. obtained by the temperature rise multiplied by the quantity of oil circulated and the specific heat of the lubricant. The losses for oil at 6,000 r.p.m. are nearly twice those obtained using water-base lubricant.

Fig. 5*b* records the motor inputs in kilowatts corresponding to loads of 60 to 154 lb. per sq. in. of projected area, and for a constant speed of 4,000 r.p.m.

An endurance test was carried out with a load of 118 lb. per sq. in. of projected area, a speed of 4,000 r.p.m., or 105 ft. per sec., for 120 hours continuously. The temperature rise of the whitemetal above

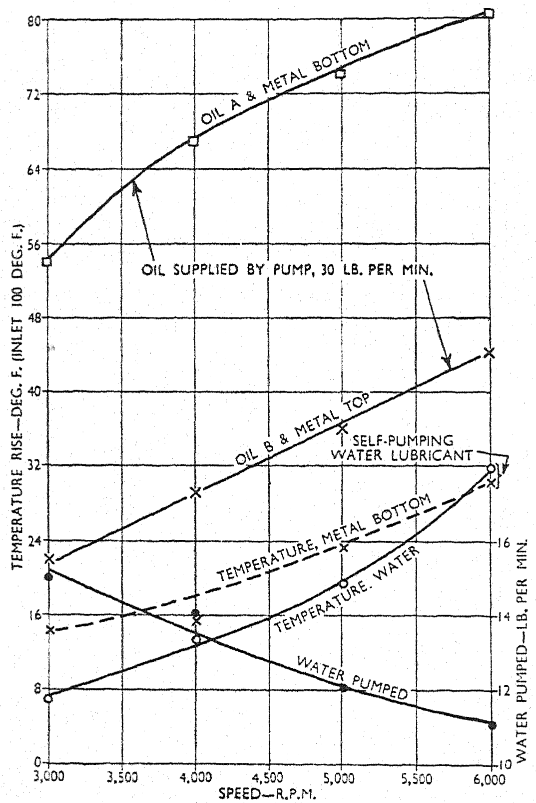


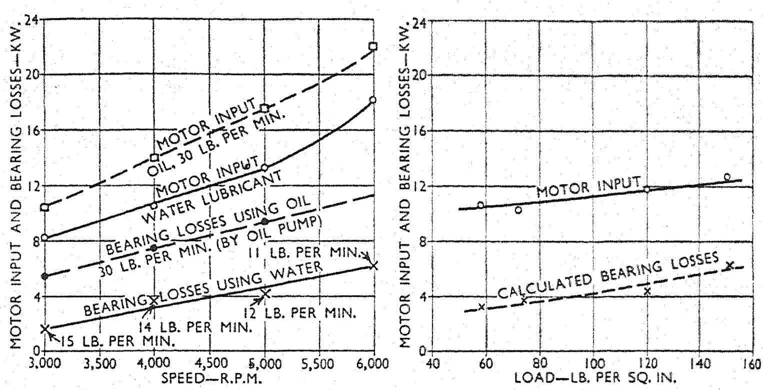
Fig. 4. Curves showing Bearing Temperatures above Lubricant Inlet and Quantity of Water Lubricant Pumped

Constant load, 71 lb. per sq. in.

100 deg. F. of the lubricant temperature at inlet was 16 deg. F. constant throughout and at the end of the run. The lubricant pumped by the journal at the commencement was 27 lb. per min., becoming 21 lb. after the first day and remaining constant at this figure until the end of the run. A gauge, located towards the top of the bearing, registered by the pumping action of the journal a maximum pressure of 10 lb.

per sq. in. After the endurance run, on opening up the bearing, the appearance of both the journal and the bearing surface was quite normal.

An interesting observation was made in connexion with the amount of lubricant pumped. It was found that the quantity was influenced more by the load per square inch of projected area on the bearing than by the speed. For example, at 57 lb. per sq. in. the lubricant pumped was 11 lb. per min. at 4,000 r.p.m., and with a load of 154 lb. per sq. in. the quantity pumped was 30 lb. per min., nearly three times the amount. The average quantity pumped during the last four days of the endurance run was 21 lb. per min. at 4,000 r.p.m. and with 118 lb. per sq. in. bearing pressure.



a

Constant load, 71 lb. per sq. in.; constant inlet temperature of lubricant, 100 deg. F.

b

Various loads; constant speed, 4,000 r.p.m., 105 ft. per sec.

Fig. 5. Motor Input and Bearing Losses

With 71 lb. per sq. in. bearing pressure, the quantity of lubricant pumped was 15, 14, 12, 11 lb. per min. for 3,000, 4,000, 5,000, and 6,000 r.p.m. respectively. At first the results were puzzling, and very careful repeat tests were made to establish the facts.

Now it has been shown that the oil film thickness varies inversely as the load and directly as the speed, or as ZN/P, where Z=the viscosity, N=revolutions per minute, and P=the pressure per unit of projected area.

The fact, therefore, of pumping more water with increased load is in accordance with anticipations, since the film thickness at the bottom of the journal would decrease with load and the clearance space at the top of the bearing would increase, allowing more fluid to flow.

It would appear that increase of speed causes a lift on the journal, with a corresponding increase of bottom film thickness. The lift would decrease the clearance space at the top of the journal, and hence counteract the tendency of the shaft to pump more liquid. At a load of 57 lb. per sq. in., the respective quantities for 3,000 and 6,000 r.p.m. were 13 and 11 lb. per min.

After concluding the tests with the water-base lubricant the same apparatus and same conditions were tried, but using oil instead of water,

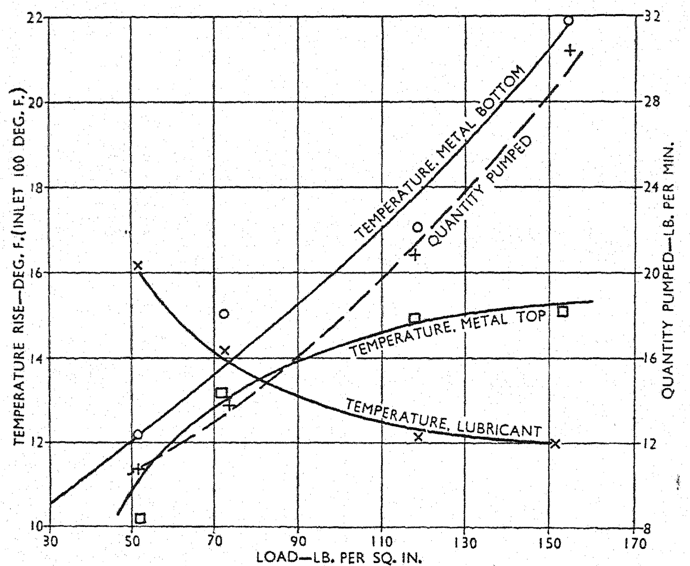


Fig. 6. Temperature Rise and Quantity of Lubricant Pumped at 4,000 r.p.m. and Various Loads

and it was found impossible to get the same results with the oil as with the water. With a load of 57 lb. per sq. in., and a speed of 4,000 r.p.m. the amount of oil pumped was 2 lb. per min., and this fell away after an hour's run to 0·4 lb. per minute. The top whitemetal temperature rose to 215 deg. F. and the bottom temperature to 207 deg. F. At this point it was shut down, since the temperature was too high to allow of satisfactory running for any length of time. Even with lower speed or increased load it was impossible to get more than 90 to 120 minutes' run before the metal temperature rose to over 200 deg. F. Satisfactory results would of course have been obtained if the oil had been put into the bearing under pressure.

The oil used had a viscosity of 280 sec. Redwood at 110 deg. F. or

about ten times the viscosity of the water lubricant. It should be noted that for the same speed the quantity pumped when the journal was working satisfactorily with the oil was about one-tenth of the water, showing that the quantity varied inversely as the viscosity of the liquid.

Fig. 6 indicates the various temperatures of lubricant, bearing metal (top and bottom), and also the amount of lubricant pumped at constant speed but with various loads. The quantity varying with the load has a curious effect on temperature, since with higher specific pressures there are bigger quantities, and the lubricant temperature rise is less for the higher loads. The bottom metal seems to show an increase in friction corresponding with increase of load. The quantity of lubricant pumped, too, seems to vary almost directly with the load, as suggested above.

The results obtained might have been expected in accordance with the physical facts and the properties of the two liquids. The two physical properties of viscosity and surface tension seem to influence results most. For example, the surface tension of oil is about 50 and that of the water 77. This would be a measure of the relative tendency to "wet" the surfaces of the journal at the bearing. The greater the tendency to "wet", the more the ultimate work done in shearing the successive layers of lubricating medium. How much chemical action influences the result is more obscure, but it is a fact that the particular water-soluble oil chosen does not oxidize appreciably, an important feature which influenced the original choice.

If the speed of rotation is high enough it would appear that the quantity pumped leads to an increase in film thickness and thus compensates for the lower viscosity of the water lubricant. In any case, whatever has been established as the criterion of ZN/P using oil, the test results with water are based upon speeds and loads accepted as standard practice for oil lubrication. Theoretically, if the speed is high enough, and the load proportionally low, the viscosity can be decreased so as to make the results equal.

It is also an accepted fact that of two oils with the same viscosity, but one containing 2 per cent of added fatty acid, the latter will show a coefficient of friction under boundary conditions half that of oil without fatty acid. It is probable, however, that the amount of dilution of the original oil would destroy any effect the 6 per cent fatty acid might have upon the coefficient of friction.

The classical experiments of the past were concerned with speeds of comparatively low peripheral velocity. The work of Goodman on journal bearing behaviour is based upon speeds below 500 r.p.m. or 12·2 ft. per sec. The speeds in the experiments described rose from a

minimum of 78 ft. per sec. to 156 ft. per sec. Further, most of the old experiments were made upon an open half-bearing, and not on a complete bearing shell, and therefore the pumping action would have a different characteristic from those of the experiments described.

No claim is made that these investigations are highly scientific, but the results are very encouraging, and it is hoped that steam turbine designers and other engineers dealing with high-temperature engines will carry on this research.

SOME APPLICATIONS OF FABRIC BEARINGS

By E. Watson Smyth*

Fundamentally, a fabric bearing is a laminated product consisting of sheets of heavy square woven duck bonded together by means of a synthetic resin of the phenol-formaldehyde type; after being submitted for a predetermined time to pressure and heat, a solid substance emerges, which has approximately the following characteristics:—

Specific gravity, 1·34 to 1·38.
Weight per cubic inch, 0·05 lb.
Tensile strength, 9,000–12,000 lb. per sq. in.
Brinell hardness No., 35–42.
Compression strength at right-angles to the laminæ, 30,000–45,000 lb. per sq. in.
Coefficient of expansion per degree C. per inch, 0·00002.
Water absorption, 24-hour test on pieces $1\frac{1}{2} \times 1\frac{1}{2} \times \frac{1}{2}$ inches thick, 0·5 per cent.
Coefficient of friction with water as lubricant, 0·016–0·023.
Safe working temperature, 100 deg. C.
Actual temperature at which material is unaffected, 150 deg. C.

Having those characteristics in mind, experiments were carried out on a wheel rolling mill of special design, the principal features of which require a main driving shaft transmitting up to 2,000 h.p., from which is driven a diagonal shaft (the bottom edging shaft) through bevel pinions, and a second diagonal shaft (the top edging shaft) driven by a separate motor transmitting 200 h.p. In view of the short life obtained from the shafts G and H and brass bearings A and C, it was decided to apply fabric to the bottom edging shaft bearing C (Fig. 1) and if results proved encouraging to apply fabric to bearings D, E on the bottom edging shaft, and to bearings A and B on the top edging shaft.

Bearing C was dealt with by trepanning the centres of two slabs of fabric material to form the bore of 12 inches diameter, and turning the outer surfaces to conform generally to section EE (Fig. 1), each half forming a complete bearing since pressure is applied only on the bottom half of the bearing (Fig. 2*a*). The top half of the bearing was made of bronze in the form of a "keep" material to give a clearance, with a suitable cavity to admit water for cooling and lubrication.

To ensure that the water flowed upwards, one round of hydraulic packing was introduced at F on Fig. 1. The packing forms a natural

* Taylor Brothers and Company, Ltd.

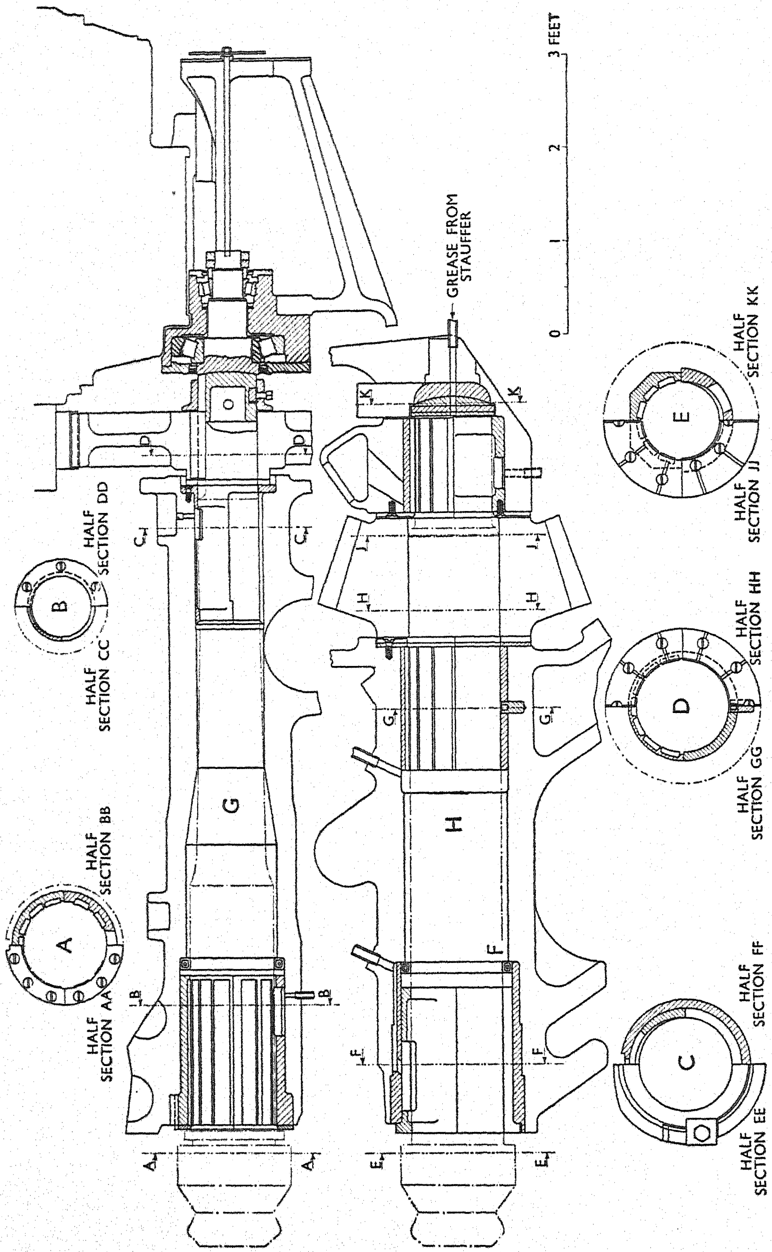
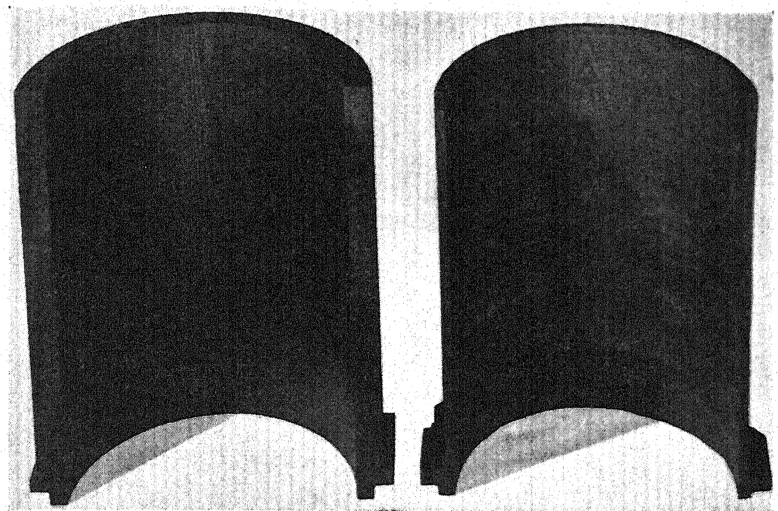


Fig. 1. Bottom Edging Shaft Bearing of a Wheel Rolling Mill

gland and the water, by flowing out at the top of the bearing, thus prevents scale from entering the bearing while the bloom is being rolled. This bearing, the most heavily loaded in the mill, is subjected to a total estimated load of 145 tons; it thus carries a load of 1,364 lb. per sq. in. on two-thirds of its projected area. The bearing is of solid fabric 19 inches long, extending over 180 deg. on a 12 inch diameter shaft. The total area over 180 deg. is $19 \times 37{\cdot}69/2 = 357{\cdot}2$ sq. in., and the bearing pressure, $145 \times 2{,}240 \times 3/357{\cdot}2 \times 2 = 1{,}364$ lb. per sq. in.

Table 1 (p. 283) shows that the bottom edging shafts gave, when



a *b*

Fig. 2. Fabric Bearing for the Bottom Edging Shaft of a Wheel Rolling Mill

running in bronze bearings, an average life of 2,300 tons or 6,750 blooms rolled. The failure of the shafts occurred always about the middle of bearing C due to fatigue flaws caused by excessive heat generated by the contact of the bronze bearing on the steel shaft. Various compositions of bronzes were tried together with various grades of steel shafts. Special lubrication was applied under pressures ranging from 100 to 2,240 lb. per sq. in. The application of a fabric bearing increased the overall life of bearing C by 2,860 per cent and that of the shaft H by 590 per cent on the basis of actual tonnage through the mill.

As a result, bearings D and E were fitted with fabric material; but the pressure per square inch being so much lower, no replacement of

these bearings has yet been found necessary, and no comparative figures are given.

A modified design of bearing was installed for the top edging-roll shaft G, consisting of a phosphor-bronze housing into which fabric strips were fitted. The strips of fabric are allowed to stand proud of

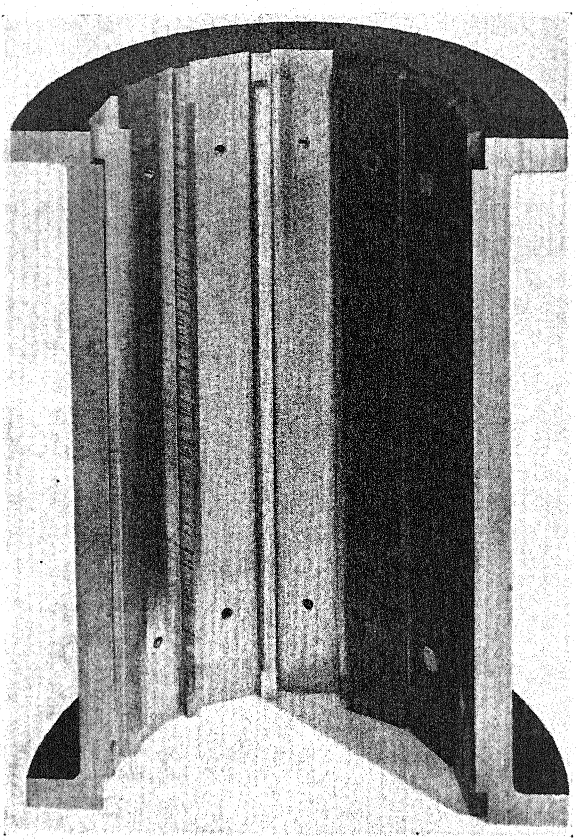


Fig. 3. Modified Design of Bearing for the Top Edging-Roll Shaft of a Wheel Rolling Mill

the housing by $\frac{3}{16}$ to $\frac{1}{4}$ inch (Fig. 3). This design reduces the effective bearing surface, but it gives a free flow of water and ensures better cooling and lubrication. Further, it reduces the cost of the bearing; and the housing should last indefinitely if the fabric strips are packed up as wear occurs. The strips can then be worn down to about $\frac{1}{2}$ inch in thickness or less according to the pressure on the bearing.

The top edging shafts G gave an average life (Table 2) when running in bronze bearings of 4,920 tons, or 14,030 blooms rolled. The failure of these shafts always occurred about the middle of bearing A due to fatigue flaws caused by excessive surface heat, as in the bottom edging roll shafts. The application of fabric bearings increased the life of the bearing by 1,390 per cent and that of the shafts by 630 per cent on actual tonnage through the mill. Bearing B has been similarly converted to fabric, but no replacement has yet been necessary. Here the bearing is of the solid construction, due to insufficient material in the steel

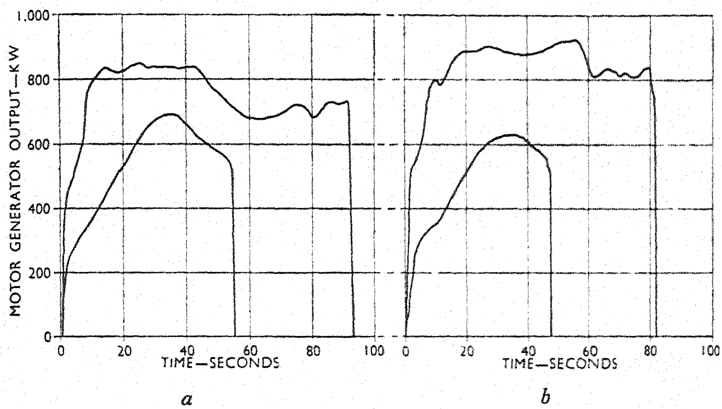


Fig. 4. Saving Effected by the Installation of Fabric Bearings

	Fig. 4*a* Upper curve	Lower curve	Reduction, percent	Fig. 4*b* Upper curve	Lower curve	Reduction, percent	Average reduction, per cent
Rolling time, seconds	94	55	41·5	82	47	42·7	42
Units consumed .	190	935	50·8	187	623	66·7	58·6
Average load . .	731	650	11·1	820	480	41·5	27·1
Peak load . .	850	690	18·8	920	630	31·5	25·4

casting for boring out to accommodate a brass housing for supporting fabric strips.

Table 3 shows that very considerable savings have been effected, amounting to 29·3 per cent increased tonnage per hour and a reduction of power consumption per ton of 39·8 per cent. Typical watt-hour meter charts show that for certain types of wheels rolled even greater savings have been effected (Fig. 4).

Fig. 4 shows the power consumption when rolling 44-inch diameter wheels, each wheel weighing 1,165 lb. Previous to the introduction

of fabric bearings the upper curves were obtained, the lower curves representing the same operation after the introduction of fabric bearings. The accompanying figures are therefore interesting. It should, however, be noted that these charts have been selected from a continuous watt-hour meter, with the smaller charts superimposed for the purpose of comparison only. The fact that one chart is of greater area than the other is accounted for by the variation in heat of the bloom coming off the forging press prior to being rolled, and the charts have been selected for comparison on this basis.

In considering the installation of fabric bearings, certain conditions must be carefully investigated, as it is possible that prevailing conditions may not lend themselves to the successful application of fabric in lieu of metal. Fabric, being a poor conductor of heat, requires an adequate supply of water to act as both a cooling medium and a lubricant. Where it is possible to provide a closed circuit the water should contain a small percentage of soluble oil. This will further reduce the surface friction, and keep the bearing surface of the shafts in a burnished condition.

Considerable wear of the fabric occurs due to rust forming during shut-down periods, unless precautions are taken to pump oil or grease into the bearing a few minutes before shutting down. With the addition of soluble oil in the service water, this will not be necessary and longer life from shafts and bearings may be expected. In order to overcome the prevalence of rust and consequent deterioration of shaft surface, experiments in coating the bearing portions of the shafts with nickel are being carried out. Unfortunately no plant is available in England for the coating of large shafts with chromium, which would give an ideal surface. Therefore, a nickel coating was tried, the nickel being deposited to a thickness of 0·020 inch per side or 0·040 inch on the diameter. This, however, was found to be unsatisfactory because the excessive pressure on the bearings caused the nickel to extrude into deep grooves. A further experiment is now under observation, with the nickel deposit reduced to 0·005 inch thick, in an endeavour to produce only a rustless surface backed by the hardness of the steel shaft beneath. It is expected that this will keep the burnished surface of the bearing on the shaft and thus further reduce the frictional effort, and effect a further saving in power.

The conversion of three tyre mills is now proceeding, and the results from the first mill were as follows: the main neck bearing on a diagonal shaft 12·75 inches in diameter carried a roll inclined 16 deg. The total load on the bearing equalled 113 tons. The top half of the top bearing is composed of seven strips of fabric $16\frac{3}{4}$ inches long $\times$ $2\frac{3}{8}$ inches wide, i.e. $39\frac{3}{4}$ sq. in. in area. The bearing

TABLE 1. WHEEL MILL: LIFE OF BOTTOM EDGING-ROLL SHAFTS AND BEARINGS

	With brass bearings	With fabric bearings	Increase, per cent
Shafts:—			
Average tonnage . .	2,300	15,840	590
Average blooms . .	6,570	41,083	525
Bearings:—			
Average tonnage . .	1,390	39,683	2,860
Average blooms . .	3,510	105,952	2,920

pressure calculated on two-thirds area, with 278·25 sq. in. total area, is $(113 \times 2{,}240 \times 3)/278 \times 2 = 1{,}365$ lb. per sq. in.

The first bearing was put into commission on 4th April 1936 and removed for strip adjustment on 5th October; it thus gave six months' service, during which time the strips had worn down to their limit. The total tyres rolled during this period amounted to 7,640 tons. A new fabric strip bearing was fitted on 5th October 1936 and has rolled to date a total tonnage of 8,615 tons of tyres. During this period the strips have been adjusted only once for wear, from which it appears that the ultimate life of this bearing may be to the order of 20,000 tons rolled. These figures are of interest when considering the saving in power and shafts. The average life of a bronze bearing on this mill was 4 weeks, with an average of 1,600 tons rolled. No figures are available for actual saving in power consumption, but the saving effected in shafts may be taken as 1 is to 3 in favour of fabric bearings.

A further experiment in design of fabric bearings of the strip type is

TABLE 2. WHEEL MILL: LIFE OF TOP EDGING-ROLL SHAFTS AND BEARINGS

	With brass bearings	With fabric bearings	Increase, per cent
Shafts:—			
Average tonnage . .	4,920	36,046	630
Average blooms . .	14,030	96,573	585
Bearings:—			
Average tonnage . .	2,470	36,954	1,390
Average blooms . .	6,330	98,950	1,465

TABLE 3. WHEEL MILL

Comparison of output and power consumption before (1935) and after (1936) the installation of fabric bearings.

		1935	1936
Weight rolled, tons		37,425·4	41,397·1
Pieces rolled		102,607	111,088
Number of 8-hour shifts		432·46	370·48
Power consumed, kW.		1,377,110	918,450
Weight rolled per shift, tons	(average)	86·7	111·7
Weight rolled per hour, tons	,,	10·8	13·96
Pieces rolled per shift	,,	237·24	299·87
Pieces rolled per hour	,,	29·65	37·48
Power consumed per shift, kW.	,,	3,184	2,479
Power consumed per hour, kW.	,,	398	309·9
Power consumed per ton, kW.	,,	36·79	22·18
Power consumed per piece, kW.	,,	13·42	8·26

	Per cent
Increased tonnage per hour	29·3
Increase in number of blooms per hour	26·4
Reduction in power consumption per hour	22·1
Reduction in power consumption per ton	39·8
Reduction in power consumption per piece	38·4

at present under observation, and results so far obtained seem to point to a possible longer life. Strips fitted in the bearings mentioned heretofore were laid with the laminæ placed horizontally, and a bearing was assembled having the laminæ of the fabric strips laid vertically.

It is proposed to use fabric strips impregnated with graphite, a material now in production, and provided this material will withstand the loads to which the bearings are submitted, improved results are looked for.

For water supply it has been found necessary to install a special centrifugal pump working at a pressure of 60 lb. per sq. in. Failure of water being fatal, a suitable warning signal is given if the pump stops owing to failure of electrical current. The importance of these points can hardly be overstressed.

BEARING PROBLEMS OF LARGE STEAM TURBINES AND GENERATORS

By C. Richard Soderberg, M.I.Mech.E.*

The present discussion is limited to journal and thrust bearings for turbines and generators in central station and industrial applications. These bearings are characterized by high rubbing speeds in combination with moderate pressures and present, therefore, ideal applications of hydrodynamic bearing theory. Forced-feed lubrication is generally used; the circulation of oil is directed principally toward the removal of generated heat.

The points of view presented in this paper apply to the practice of the firm with which the author is associated and do not necessarily represent universal practice in the United States.†

The bearings supporting the generator rotor are usually made identical with those of the turbine. It is sufficient, therefore, to illustrate the discussion by Fig. 1, which shows a longitudinal section through the front pedestal of a steam turbine, and provides a typical example of the journal and thrust bearings under discussion.

Journal Bearings. The shell of the journal bearing (Fig. 1) is usually of cast steel in order to ensure good bonding with the Babbitt metal. Accurate alignment dictates the use of aligning keys, which in the lower half are placed at 45 deg. from the vertical centre line. The outer surface of these keys is usually spherical, although for smaller turbines cylindrical fits are also common. The oil is admitted through one of the keys in the bottom half and enters the annular space of the top half. From here, a small portion goes into the oil film proper; the remainder flows axially through grooves in the Babbitt metal, removing heat from the shaft by mixing with the oil which has been heated in the oil film. The oil is discharged through separate pipes which carry the flow to the lower portion of the pedestal to avoid foaming. Orifice plugs in these pipes permit the flow of oil to be adjusted. The sealing is designed to give a minimum of end leakage and assurance against vapour leakage to the atmosphere. The load-carrying part of the bearing is confined to an arc of 90 to 120 deg. disposed centrally about the vertical centre line. This active arc usually has a radius of curvature 2/1,000 to 4/1,000 greater than the

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* The Westinghouse Electric and Manufacturing Company, Philadelphia.

† A paper of similar scope was presented before The Institution of Mechanical Engineers in 1929 by Francis Hodgkinson. (Proc. I.Mech.E., 1929, p. 843.)

radius of the journal itself. The higher values are used with the highest steam temperatures, where a possible difference of temperature between the shaft and the bearing will tend to reduce this clearance. The bearings are intended to operate with an actual clearance of about (2/1,000)d. This type of bearing has been in general use for several

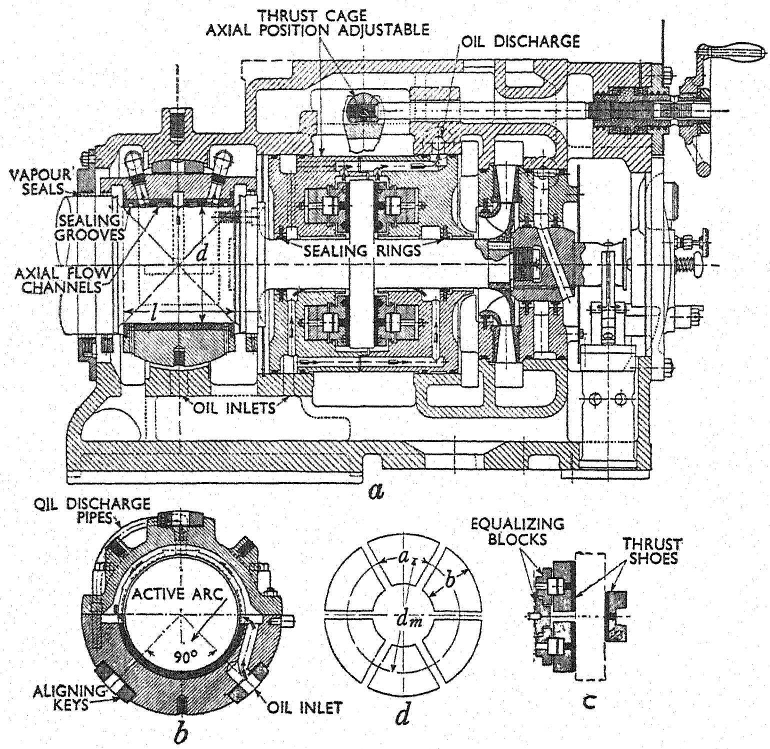

Fig. 1. Section through Front Pedestal and Arrangement of Bearings of Typical Steam Turbine

years and is considered a satisfactory solution of the journal bearidg problem. It has been applied successfully to the following conditions:—

Speed of rotation, r.p.m.	Maximum diameter, inches	Peripheral speed, ft. per sec.	Maximum pressure, lb. per sq. in.
1,800	24	189	185
3,600	12		

There is at present no urgent incentive to exceed 189 ft. per sec. in speed, but there is no fundamental reason why it cannot be exceeded. The pressure has occasionally been increased to 200 lb. per sq. in. without serious trouble. This bearing is manufactured in two lengths: $l=d$, and $l=1{\cdot}5d$.

Journal bearings of this type, operating under the conditions usually encountered in turbo-generators, no longer present serious problems as regards hydrodynamic functioning in the steady state. The solutions are fully covered by existing literature (cf. Howarth 1932, 1934, 1935; Kingsbury 1932), and only a few of the major aspects will be reviewed in this paper. The following notation is used:—

d	Journal diameter, inches.
δ	Radial clearance, inches.
$h=\delta-\rho$	Minimum oil thickness, inches.
l	Length of bearing, inches.
$m=2\delta/d$	Clearance ratio.
N	Speed of rotation, revolutions per minute.
p	Pressure, pounds per square inch.
P	Mechanical loss, kilowatts
ρ	Eccentricity of shaft centre, inches.
z	Viscosity of oil at discharge, centipoises.

The operation in the steady state is characterized by equilibrium between the external forces and the hydrodynamic reactions from the oil film. The nature of this equilibrium is illustrated in Fig. 2*a*. O is the centre of the active journal surface and O_1 the position of the shaft centre under a certain combination of pressure p, speed N, viscosity z, and clearance ratio m. For each value of the clearance ratio m, the circumstances are determined by the parameter zN/p, which plays an important role in this problem, as well as in the thrust bearing problem. As this parameter is varied, the shaft centre O_1 assumes different positions on the equilibrium curve OO_1O_2 (Fig. 2*b*). This curve is the locus of all shaft positions which produce hydrodynamic forces having a vertical resultant. Its shape is not entirely independent of clearance ratio, supporting angle, etc. The form shown in Fig. 2*b* is representative for typical bearings. The conditions which determine the ratio ρ/δ are shown in Fig. 2*c*. This result makes it possible to predict the minimum oil film thickness

$$h=\delta-\rho=\delta(1-\rho/\delta) \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

This is significant in establishing the limit of pressure, but cannot be used as an exact criterion, because of influences such as finish, compressibility of Babbitt metal, etc., not to mention the approximate

nature of the theory. It gives a qualitative insight into the way the bearing works, however, which is of considerable value.

The mechanical losses of such bearings cannot be based entirely

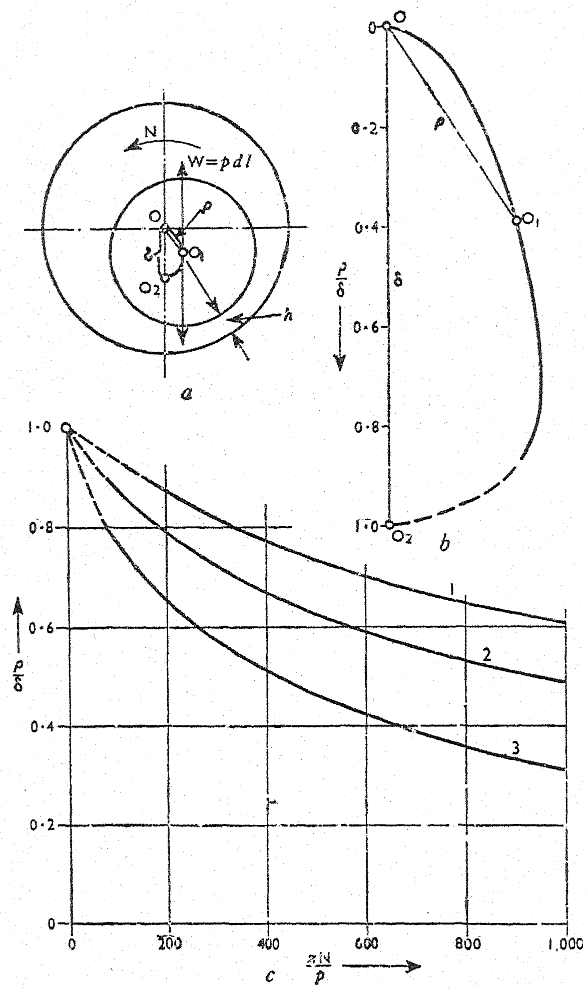

Fig. 2. To Illustrate the Functioning of Journal Bearings

upon theoretical considerations, because of the losses in the top half, side leakage, etc. Exact empirical formulæ are difficult to obtain, because of the difficulty of making measurements of the exact loss.

The following formula, based on theoretical reasoning, has been found to duplicate test results quite satisfactorily.*

$$P=k_1 z d^2 l N^2\left(1+\frac{k_2}{zN/p}\right) \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

k_1 and k_2 are constants, which depend upon clearance, finish, etc. For journal bearings of the type shown in Fig. 1 with 90 deg. active arc, and $m=2/1{,}000$, several series of tests have been duplicated by $k_1=0{\cdot}75\times10^{-10}$ and $k_2=100$. The second term of the parenthesis is of major importance only for small values of zN/p (low speeds). For conditions normally obtaining in constant-speed turbines, this term may be dropped and the equation for loss written

$$P=0{\cdot}12z\frac{d^2l}{1{,}000}\left(\frac{N}{1{,}000}\right)^2 \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

The flow of oil required to maintain the bearing temperature at a certain limit is theoretically based upon this loss. However, radiation and conduction of heat from the steam require oil flows of two to three times this amount.

Thrust Bearings. The type of thrust bearing in general use (Fig. 1) is of the Kingsbury type with self-aligning shoes. The number of shoes varies from six to twelve, and may be as high as twenty for very large bearings. It is undesirable to have more than twelve shoes because of friction in the self-aligning blocks. The shoes are usually proportioned so that $a/b=1$, but satisfactory operation has been obtained for $0{\cdot}75<a/b<1{\cdot}25$. The shoes are supported at the centre of gravity of the bearing surface. Lubricating oil is admitted at the inner radius of the shoes. Rounding of the shoes at the inlet edge is important. The shoes are spaced so as to give a ratio of shoe area to gross annular area of about 0·85. This type of thrust bearing has been applied successfully to the following operating conditions:—

Speed, r.p.m.	Number of shoes	Mean diameter, inches	Peripheral speed at outer edge of shoe, ft. per sec.	Maximum pressure used successfully, lb. per sq. in.	Pressure normally allowed, lb. per sq. in.
1,800	12	13·75	132	500	250
3,600	6	6·72	143	400	400

* Suggested by H. Semar, of the Westinghouse Electric and Manufacturing Company, on the basis of a series of gear tests.

The journal bearings which are contiguous to the thrust bearings in conventional steam turbines carry a comparatively small load. There is no inconsistency, therefore, in the fact that the limit of peripheral speeds for thrust bearings is lower than that of journal bearings.

The hydrodynamical behaviour of this class of bearings is also well known, and only a few of the practical aspects will be given in the following. The following notation is used:—

a Circumferential dimension of shoe (at d_m), inches.
β Dimensionless ratio.
b Radial width of shoe, inches.
d_m Mean diameter, inches.
h Mean oil film thickness, inches.
k Constants.
μ Coefficient of friction.
ν Number of shoes.
N Speed of rotation, revolutions per minute.
p Mean pressure, pounds per square inch.
P Mechanical loss, kilowatts
z Viscosity of oil at discharge, centipoises.

In the functioning of the thrust bearing, the dimensionless ratio $zN\nu/p$ is important. The mean oil-film thickness, the coefficient of friction, and the mechanical loss are given by the equations

$$\frac{h}{a}=k_h\sqrt{zN\nu/p} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

$$\mu=k_\mu\sqrt{zN\nu/p} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

$$P=k_p\sqrt{zpN^3\nu}d_m{}^2b \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The dimensionless constants k_μ, k_h, and k_p have the approximate values

$$k_h=25\times10^{-6};\quad k_\mu=152\times10^{-6};\quad k_p=2{\cdot}4\times10^{-9} \quad . \quad (7)$$

In most instances the turbine thrust bearings are used in pairs (Fig. 1), particularly where the turbine rotor is shifted axially to obtain varying clearances. The thrust bearings are then set with a total axial clearance h, generally of the order of 0·010 to 0·015 inch. These clearances are small enough to produce a perceptible pressure reaction in the opposite direction on the idling thrust bearing, so that the main thrust bearing has to carry a slightly larger load. Introducing the ratio

$$\beta=\frac{k_h a}{h}\sqrt{\frac{zN\nu}{p}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

when p is the pressure due to the external thrust load, it is found that the two thrust bearings actually carry the loads

$$p_1=(1+\beta^2)p;\quad p_2=\beta^2 p \quad . \quad . \quad . \quad . \quad . \quad . \quad (9)$$

Their combined loss is

$$P=k_p\sqrt{zpN^3\nu}\,d_m{}^2b(\beta+\sqrt{1+\beta^2}) \quad . \quad . \quad . \quad . \quad (10)^*$$

Double thrust bearings of this kind frequently operate without external thrust load. The pressure generated when the total clearance is h is easily found to be

$$p_o=\frac{4k_h{}^2a^2zN\nu}{h^2} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (11)$$

The combined mechanical loss is

$$P=k_p\frac{4k_ha}{h}zN^2\nu d_m{}^2b \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (12)$$

The circulation of oil required to maintain the oil temperature must be increased beyond the value corresponding to this loss in order to allow for radiated and conducted heat. In addition, the losses must be corrected for the friction loss against the outside of the thrust collar and the inside shafting.

Oiling System. In a modern steam turbine lubrication is only one of the functions of the oiling system. The control valves and the stop valve require forces of actuation which may reach the order of 10 tons, so that hydraulic operation of the valves is a necessity. There must be auxiliary pumps with pressure regulation, and when the main pump is mounted directly on the turbine shaft there is usually an ejector to maintain the suction pressure. Operation of the turning gear may require still other pumps with regulators. The motion of the control valves may be a function of exhaust or extraction pressures as well as of speed. The method of cooling the generators with hydrogen has brought about the need for separate circuits of the oiling system for sealing purposes. These many functions have complicated the oiling systems of modern turbines to the point where they represent important separate plants.

The troubles which have occurred with the oil relate chiefly to sludging and corrosion. Recently the newer types of solvent-refined oils have been reported to accentuate corrosion troubles. Copper is known to intensify sludging through catalytic action, but no definite

* This method involves an approximation which, however, is negligible for values of β less than 0·25.

policy of excluding copper from the oiling system has crystallized. For example, brass tubes for oil coolers are quite common. The viscosities range from 140 to 250 sec. Saybolt at 100 deg. F., and from 40 to 50 sec. Saybolt at 210 deg. F. Oiling systems which include gears use heavier oil. The capacity of the oil tank is now usually based on the assumption of a circulation period of 5 minutes. The maximum temperature of the oil is usually limited to 140 deg. F., but may go to 160 deg. F.

Lead-base Babbitt metal has been practically discontinued, so that troubles due to the formation of oleate of lead are no longer of importance. The tin-base Babbitt metal in general use has the approximate percentage composition: tin, 85; copper, 5; antimony, 10; lead, less than 0·35. Bearing shells and shoes are of steel and are always tinned before Babbitting.

Problems Incidental to High Steam Temperatures

Heating the Oil by Conduction and Radiation. The conditions encountered at the inlet end of modern turbines have been much accentuated by advances in steam temperatures. The temperature of the cylinder itself is usually very little below that of the inlet steam, or about 900 deg. F. in modern machines. About 1 foot away from this point there is a Babbitted bearing. The Babbitt metal of the bearing will melt at about 500 deg. F., and lubricating oil will burn at around 550 deg. F. The relative success of this arrangement is, of course, associated with the fact that the rotor can be cooled effectively enough to keep its temperature far below these critical temperatures. In the typical machine there are two major means for withdrawal of heat: the water gland and the journal bearing. This function of cooling on the part of the water gland is one of the reasons for its continued use on high-temperature turbines. The dissipation of heat to the lubricating oil from the shaft is an important matter in high-temperature turbines, which must be considered in the design of the oiling system, and particularly in proportioning the oil coolers. Heat conduction through the shaft is not the only source, however. It is probable that radiation from the hot steam surfaces and absorption by the bearing pedestal surface also play an important role. For turbines of normal construction, operating at steam temperatures of 825 deg. F. and above, the amount of heat which is conducted and radiated into the oiling system is from two to three times as large as the heat generated by friction in the bearings.

Fire Hazard. A few years ago a series of destructive oil fires occurred in central station turbines in the United States. With some

exceptions, the fires did not occur in units with high steam temperatures, nor were they confined to any particular turbine or station arrangement. In some instances the fires could have been prevented by better methods of operation. These accidents led to a wholesome attention to the problems of fire hazard incidental to the oiling system, which has influenced the arrangements used to-day in several respects. The most important conclusions drawn from a study of the circumstances which attended these oil fires were:—

(1) The fires were caused by failures of pipe fittings, gaskets, and similar details, causing oil to pour over unprotected hot steam parts.
(2) The early circumstances of most fires caused the operators to regard them as more or less trivial incidents. The really serious aspects were brought on by the operator's reluctance to stop the machine. In the latter stages of the fires, the severe destruction was due to the burning of very large amounts of lubricating oil.

As a result of these fires, efforts were applied towards complete elimination of the fire hazard, and the use of a non-inflammable fluid for the operation of all steam valves was a natural approach. The only fluid which received serious consideration in this connexion was "Aroclor" (chlorinated diphenyl). Several important units were equipped with dual oiling systems of this kind, notably the 165,000 kW. turbine at the Richmond Station of the Philadelphia Electric Company. These have operated quite satisfactorily. Even though the existing Aroclor systems in the United States have functioned without difficulties, the development of the last two years has been directed towards solutions based on the use of inflammable oil for valve operation as well as for lubrication.

Since the front pedestal must always be supplied with oil for lubrication, it has seemed logical to extend this pedestal into an oiltight casing which houses, in addition to the bearings, the main pump and the governor. This location of the governor is postulated on the possibility of actuating all control valves from a single mechanism, which is the case for the Curtis reaction turbine equipped with the bar-lift steam chest. The use of a centrifugal oil pump, mounted directly on the turbine shaft, also favours this arrangement. Its value as a means to reduce fire hazard lies in the possibility of assigning a suitable location for the oil tank, even to the extent of housing it in a separate fireproof compartment.

The only other serious element of fire hazard is the stop valve. This consideration has brought about a new type of inverted oil-operated

stop valve, in which the oil-operated mechanism is located underneath the valve. This is usually located below the floor and connected with the turbine through one or two steam pipes. In this manner it has been possible to devise an oiling system in which probable failures of oil piping will not lead to flooding of hot steam parts. Not the least important gain is the reduction in the number of oil pipes surrounding the front end of the turbine. Of equal or even greater importance are improvements in details of the oiling system. Seamless tubing is used almost without exception, and bolted joints are replaced by welding wherever possible. The remaining bolting is usually of 250 lb. per sq. in. standard, and screwed joints are not permitted. The flanges are of the weld-neck type, and the welding is given very careful inspection. This revision of major and minor details, accompanied by improved housekeeping in the central stations, has materially reduced fire hazard in modern turbines.

Problems of Operation

Slow Rolling. In the operation of large high-temperature turbines the greatest risks occur during starting and stopping. The most dangerous circumstances are encountered when a turbine is permitted to remain stationary for any length of time immediately after a shut-down. The possible distortions of the rotor may make it impossible to start again for several hours. As a result of this situation the turning gear has become an indispensable auxiliary for steam turbines. Large turbines are never permitted to stand still but are put on the turning gear as soon as they are shut down. Experience obtained with modern large high-temperature machines indicates that frequent starts and stops are simply not possible without the turning gear. A separate motor-driven oil pump, designed to pass the limited amount of oil required for turning gear operation, is usually supplied for these large machines.

Vibration and Similar Phenomena. In recent years there has been a great deal of discussion about phenomena of dynamical instability of the oil film, and many serious vibration troubles have been diagnosed as examples of such phenomena. Most of the theoretical questions raised have been cleared up and there is already an extensive amount of literature on the subject (Newkirk 1925; Stodola 1925; Hummel 1926; Robertson 1933, 1935).

The phenomenon referred to as "oil whip", in which the vibration frequency is half the rotational frequency, is rare in steam turbines. It is usually associated with high values of zN/p and with types of bearings

in which the amount of oil supplied is apt to be insufficient for complete lubrication. It is particularly apt to occur where the machine operates at twice its critical speed. The experience available to the author indicates that for bearings of the type described it is never serious.

There are, nevertheless, many cases of vibration troubles which are accentuated and perhaps caused by the elastic properties of the oil film. Referring to Fig. 2*b*, it is evident that the value of ρ represents the elastic deflexion of the oil film under a vertical force equal to gravity. If, due to unbalance or other causes, there is an alternating force, it will cause the shaft centre to describe some closed curve around this equilibrium position. It is evident that this loop is relatively more extensive the higher the value of zN/p. When the loop involves displacements of the same order of magnitude as ρ, the motion will lose its regular character. Troubles which can be diagnosed on this basis have been encountered in high-speed machines. The oil film is not the original cause, however, but merely the means of magnifying the motion. The fundamental cause is eccentric motion of the journal, produced either by inaccuracies of machining, a slight bend in the shaft, or ordinary unbalance. This experience indicates that the advance toward higher values of zN/p will justify efforts to prevent the introduction of this class of errors. In severe cases, the phenomenon has caused pitting and flaking of the Babbitt metal. This is a particularly obnoxious form of trouble, because it may require months of operation to be apparent. So far most cases have responded to balancing or to elimination of other errors.

Vibration troubles in thrust bearings are rare, but have occurred in a few cases. In one case the trouble was diagnosed as self-excited vibration under the influence of thrust forces. The trouble, diagnosed by J. G. Baker (Den Hartog 1934), was eliminated by an expansion chamber in the equilibrium connexion. It has not been encountered on more than one unit, in which it was brought about by poor design of the sealing details and the support of the thrust bearing.

References

Den Hartog. 1934 "Mechanical Vibration", McGraw Hill, London.

Howarth, H. A. S. 1925 Trans. A.S.M.E., vol. 47, p. 1073, "Graphical Analysis of Journal Bearing Lubrication".

1934 Trans. A.S.M.E., vol. 56, MSP–56–2, "Speeds, Clearances and Lubrication of Oil Film Bearings".

1935 Trans. A.S.M.E., vol. 57, MSP–57–2, "Loading and Friction of Thrust and Journal Bearings with Perfect Lubrication".

HUMMEL. 1926 "Kritische Drehzahlen als Folge der Nachgiebigkeit des Schmiermittels in Lager," VDI-Verlag.

KINGSBURY, A. 1932 Trans. A.S.M.E., vol. 54, RP–54–7, "Optimum Conditions in Journal Bearings".

NEWKIRK, B. L., and TAYLOR. 1925 *General Electric Review*, August, p. 559, "Shaft Whipping due to Oil Action in Journal Bearings".

ROBERTSON, D. 1933 *Phil. Mag.*, January, p. 113; 1935 Proc. I.Mech.E., vol. 131, p. 513, "Hysteretic Influences on the Whirling of Rotors".

STODOLA, A. 1925 *Schweizerische Bauzeitung*, vol. 85, p. 265.

THE EARLIER HISTORY OF THE HYDRODYNAMIC THEORY OF LUBRICATION FRICTION

By Professor A. Sommerfeld, F.R.S.*

During the last century the belief of technicians in Coulomb's laws for dry friction was so general that it even permeated the problem of lubricating friction. The frictional moment between journal and brass was for that reason formulated as

$$M=frP \qquad (1)$$

where P is the total load carried by the bearing, r the radius of the journal, and f a dimensionless value. Whereas with dry friction, according to Coulomb, the moment would be independent of velocity and pressure, its actual dependence on velocity and pressure still requires in every case experimental investigation. N. Petroff was the first to relate bearing friction with the viscosity of the lubricant, and in his book, "Neue Theorie der Reibung" he gives instead of equation (1) the formula

$$M=\mu rA(U/d) \qquad (2)$$

where μ is the coefficient of viscosity, A the surface wetted by the lubricant, U the peripheral speed of the journal, and d the clearance between brass and shaft when the latter is placed centrally.

In 1886 Osborne Reynolds, without any knowledge of the work of Petroff, published his famous treatise which gave a deeper insight into the hydrodynamic theory of lubrication. Reynolds recognized that the journal cannot take up a central position in the bearing, but must so find its position according to speed and load that the conditions for the equilibrium of forces (the resultant and moment of all the forces equal zero) are satisfied. At high speeds the eccentricity of the shaft in the bearing decreases, but at low speeds it increases, the shaft then tending to move forwards in the direction of rotation. The lubricant then has to force its way through the narrow slit between journal and bearing, so that the friction is increased.

Both the inertia of the lubricant and its specific gravity can be neglected, and the problem can be treated as two-dimensional, i.e. the journal and brasses are considered to be of infinite length. Despite this simplification Reynolds's equation leads to complex and obscure expressions, for the reason that Reynolds dealt with a half-brass, particularly in view of Beauchamp Tower's researches in 1883 and 1884. The

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theory, however, becomes much simpler, as the author contended in 1904, when a fully enclosed brass with film lubrication is considered, as in bearings of high-speed electrical machines. Complete integration can then be carried out, and the eccentric position of the journal in the brasses can be calculated with accuracy. Let e be the eccentricity of the journal, and d, as in equation (2), the difference between the radii of brass and shaft, then e/d is a simple function of the quantity

$$Z=(r/d)^2\mu U/P \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

P represents the bearing load per unit length of axis, and the remaining notation is as in equation (2). The appearance of the quantity Z, which is a pure number (dimension 1), is the law of similarity of the hydrodynamic theory of friction.

If $Z=0$, then $e/d=1$; the journal rests directly on the brass; if $Z=\infty$, then $e/d=0$; the journal runs centrally. The position of e is always perpendicular to P (horizontally when P is vertical) and arises from P by turning 90 deg. in the direction of rotation, corresponding to Reynolds's "moving forwards".

The frictional moment will in general be given by

$$M=2\pi\mu r^2(U/d)F(Z) \quad . \quad . \quad . \quad . \quad . \quad . \quad (4)$$

The first factors are identical with Petroff's formula (2) (in the case under consideration the wetted surface A will, if it is calculated like P and M on unit length of the axis, equal $2\pi r$). As to the factor F(Z) the author limits the discussion in this paper to the two limiting cases

$$Z=0, \text{ that is, } U=0 \text{ or } P=\infty$$
$$\text{and } Z=\infty \quad ,, \quad ,, \quad U=\infty \text{ or } P=0.$$

In the first limiting case we have $F(Z)=1/2\pi Z$, and from (4) and (3)

$$M=(d/r)rP \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

so that M becomes proportional to P; equation (5) is equivalent to equation (1) with

$$f=d/r \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The coefficient of friction defined in this way becomes independent of P and U, as it should according to Coulomb.

In the second limiting case we have $F(Z)=1$. Equation (4) then becomes identical with Petroff's formula (2); the frictional moment becomes independent of P and proportional to the velocity U, which is at variance with Coulomb. The coefficient of friction defined according to Coulomb will now be proportional to Z. Thus,

$$f=M/rP=2\pi(r/d)(\mu U/P)=2\pi(d/r)Z \quad . \quad . \quad . \quad . \quad (7)$$

Between these two limiting cases there exists a characteristic transitional value

$$Z = Z_{\text{min}} = 5/24\pi \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (8)$$

for which M is a minimum.

In Fig. 1, Z is the abscissa and $(r/d)f$ the ordinate. According to theory, all points representing actual cases of friction should lie on the full line. For Z=0 the curve touches the horizontal line labelled

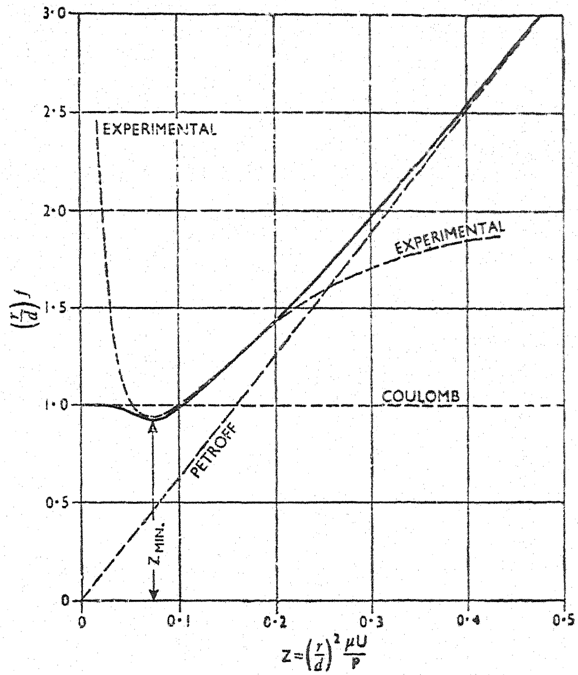


Fig. 1. Illustrating the Law of Similarity of Lubricational Friction

"Coulomb" (equation (6)); it attains for $Z = Z_{\text{min}}$ (equation (8)) a minimum which lies slightly below the Coulomb horizontal line; for increasing values of Z, it approaches the straight line labelled "Petroff" (equation (7)). This curve illustrates the law of similarity of lubricational friction, mentioned previously.

What conclusion can be drawn from the experimental data? (Reference is made chiefly to the work of Stribeck (1902), later supplemented by the work of Biel (1920)). The answer is given by the curve

marked "experimental" in Fig. 1. Near the minimum the law of similarity is confirmed quantitatively, as it is by the rising branch of the curve. When Z is greater, the experimental curve separates from the theoretical curve, continuing less steeply. This can be explained qualitatively by the rise in temperature, and the consequently decreased viscosity of the lubricant, whereby, in agreement with the Petroff equation (2), the increase of the frictional moment is reduced.

Of more importance is the deviation of the experimental and theoretical curves when Z is small. The observation points now rise steeply to a high multiple of the theoretical coefficient of friction at rest. Even the law of similarity fails, in so far as the separation of the experimental from the theoretical curve occurs not for a definite value of Z but for various values of Z according to the construction and load of the bearing. The reason for this discrepancy lies, first of all, in the fact that, when there is considerable eccentricity of the journal the theory must make allowance for negative pressures, which should occur immediately behind the position where journal and brass are most close together and which in practice lead to shearing of the layer of lubricant. Another probable explanation is that the theory deals with the general viscosity constants even for the thinnest layers, and neglects the molecular structure of the oil which, according to Langmuir, probably plays a part.

The author's work was published (1904) in the *Zeitschrift für Mathematik und Physik*, now discontinued. In vol. 52 (1905) of that journal, A. G. M. Michell describes, in direct connexion with the author's work, the theoretical basis underlying the conception of his well-known thrust bearing. Those two papers were reprinted together with an extract from Petroff's book and with the main part of Reynolds's treatise in one of "Ostwald's Klassikern der exakten Wissenschaften" (1927), together with a later review of the theory due to the author (1921). More recent literature on the subject is, unfortunately, unknown to the writer, though, according to Foote (1937), papers and a book by M. D. Hersey should be specially noted.

In conclusion, attention may be drawn to a general question, previously raised by the author (Klein and Sommerfeld), namely : Is it possible to attain a closer comprehension of dry friction by considering the air remaining between the two rubbing surfaces as a kind of lubricant ? The author learns that Ragnar Holm* has propounded the same question as a result of exhaustive researches with polished metal surfaces (1936) and of considerations relating to the theory set out in this paper.

* Written communication to the author.

References

BIEL, C. 1920 Z.V.D.I., vol. 64, p. 449.

FOOTE, P. 1937 Jl. Applied Physics, vol. 8, p. 19.

HOLM, RAGNAR. 1936 Wissenschaftliche Veröffentlichungen aus den Siemens-Werke, vol. 15, p. 122.

KLEIN, F., and SOMMERFELD, A. "Theorie des Kreisels", Teubner, Leipzig, chapter 7, p. 546.

MICHELL, A. G. M. 1905 *Zeitschrift für Mathematik und Physik*, vol. 52, p. 123.

OSTWALD'S Klassikern der exakten Wissenschaften, 1927, No. 218 (edited by L. Hopf), Akademische Verlags-Gesellschaft, Leipzig.

PETROFF, N. "Neue Theorie der Reibung."

REYNOLDS, O. 1886 Phil. Trans. Roy. Soc., vol. 177, part 1, pp. 157–234.

SOMMERFELD, A. 1904 *Zeitschrift für Mathematik und Physik*, vol. 50.
1921 *Zeitschrift für technische Physik*, vol. 2, Nos. 3 and 4.

STRIBECK, R. 1902 Z.V.D.I., vol. 46, p. 341.

TOWER, B. 1883 Proc. I.Mech.E., p. 632.
1884 Proc. I.Mech.E., p. 29.

LUBRICATION AS APPLIED TO LOCOMOTIVE JOURNALS.

By W. A. Stanier, M.I.Mech.E. (*Vice-President*)*

Theoretical Aspects. The lubrication of axle journal bearings on locomotives is a modification of that of a rotating shaft. The most complex case is that of coupled axle bearings and the following applies more particularly to that case.

The ideal state, i.e. viscous film lubrication with the minimum viscosity commensurate with the load and speed, would only be possible on a train which never stopped. Assuming the heat generated in the ideal bearing to be entirely due to the shearing of the oil film, the viscosity would tend to accommodate itself to the speed and the resultant equilibrium temperature would eventually be acquired by the whole mass of the bearing metal. On slackening speed or stopping, the oil could not accommodate itself to the new conditions by thickening up again because of the heat held in the bearing. It is during these all-important intervals that one has to rely more or less entirely on boundary and greasy lubrication and a great load is imposed on the whitemetal bearing surface. This can be lessened by using a thicker oil, the objection to which lies in the wastage of energy in shearing the oil film. Now since the whole of this energy appears as heat, the temperature of the box can be taken as its measure and it is better to lose power in this way than by the rubbing of the bearing surfaces. Thus, if an oil of double the viscosity gave a bearing temperature similar to that given by a thinner oil it could be argued that since it is known that more work has been expended on shearing the thicker oil, the difference in respect of the thinner oil can only be attributed to heat generated by rubbing friction, which it is the object of lubrication to eliminate. However, the work expended on viscous shearing in the coupled axlebox is a very small portion of the total power output of the engine.

After standing at rest for a time the oil film at the crown of the bearing becomes so attenuated that it can be assumed that purely boundary conditions prevail. On restarting, therefore, the bearing surface is severely taxed for a short time until the film is re-established. The thinner the oil the longer is this time, the acceleration being the same. In the coupled axlebox of a locomotive these adverse starting conditions are considerably improved because the load on the bearing is transferred

* London, Midland and Scottish Railway Company.

from the dead vertical at the instant of starting by the horizontal thrust of the crank or connecting rod.

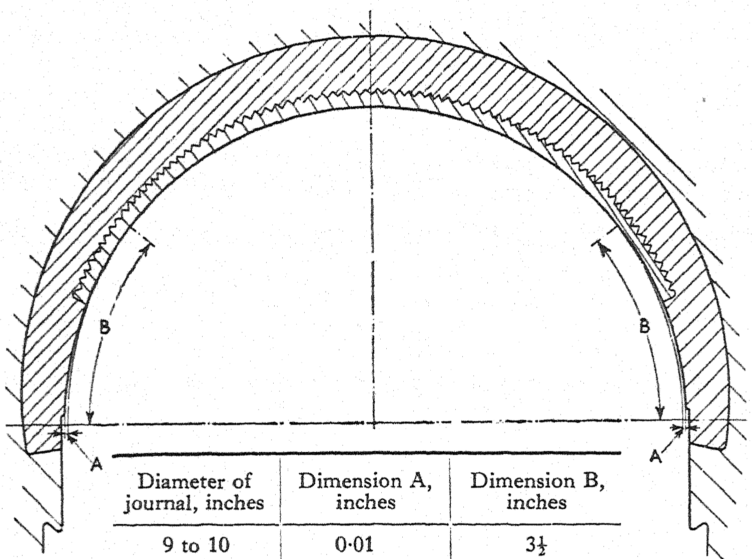
In axleboxes for carrying wheels, there is no horizontal thrust to transfer the load from the vertical to alternate quadrants as in a coupled axlebox, and conditions at starting must be correspondingly more severe, but once motion has started, everything is favourable to the formation of a true viscous film.

Experimental Work. Many experiments have been made to determine the maximum and minimum temperatures of an axle bearing under running conditions and a few to determine the temperature more or less continuously. All the experiments suffer because the temperatures were measured, not in the bearing, but in the brass at an appreciable distance ($\frac{1}{8}-\frac{1}{2}$ inch) from the surface. The results show very little difference in the temperature whether the thermometer or thermocouple is $\frac{1}{8}$ inch or even $\frac{1}{2}$ inch away from the bearing surface. Recently, temperature readings of four coupled axle brasses were taken over several hundred miles at two-minute intervals by means of thermocouples $\frac{1}{8}$ inch from the bearing surface. These differed for each axle, but were all in the neighbourhood of 60 deg. C. (140 deg. F.) while the locomotive was running at about 60 m.p.h. The temperature of each brass rose with increase of speed, but the variation from brass to brass was often greater than from speed to speed. Thus in one case, the temperatures of the four brasses at 60 m.p.h. varied from 50 to 62 deg. C. (122–143·6 deg. F.) whilst an additional 10 m.p.h. added only 2–3 deg. C. (3·6–5·4 deg. F.) to the temperature. Such results suggest that the lubrication is of the viscous film kind when the locomotive is running at speed. The results seem also to indicate that conditions differ from bearing to bearing; this is to be expected, since the dissipation of heat from the brasses depends on the proximity and the temperature of other surfaces, the direction and force of the wind, and on other possible variables such as load and alignment. It should be stated that the temperatures were measured on an engine in excellent running condition: further, during the running-in period, temperatures up to 80 deg C. (176 deg F.) were observed for a brass which subsequently gave temperatures of about 55 deg. C. (about 130 deg. F.). In an experimental bearing, temperatures have been taken as near as $\frac{1}{32}$ inch to the bearing surface. These were only very slightly higher than those taken $\frac{1}{8}$ inch away: the actual temperatures are not given, as the difference did not greatly exceed the possible experimental error and in any case was not of magnitude sufficient to show the temperature of the actual bearing surface.

Practical Aspects. In the modern locomotive the bronze axlebox

for coupled wheels has given place to a cast steel box with a pressed-in brass, lined with a whitemetal bearing surface to within $\frac{3}{4}$ inch of the horizontal centre line. In addition to its lower initial cost the cast steel box gives greater strength and reliability. Depending on its diameter, the box is bored up to 0·020 inch larger than the diameter of the journal in order to provide a lead-in for the oil from the spring pad in the keep (Fig. 1).

A worsted spring pad is provided in the keep, and in addition direct



Diameter of journal, inches	Dimension A, inches	Dimension B, inches
9 to 10	0·01	$3\frac{1}{2}$
8 to 9	0·0078	$2\frac{3}{4}$
7 to 8	0·0058	2
Below 7	0·0043	$1\frac{1}{2}$

Fig. 1. Clearance of Brasses for Locomotive Axleboxes

lubrication to the brass is supplied either by means of an auxiliary oil box and trimming or a mechanical lubricator. Attention must be paid to the quality of the worsted in the pad since the siphoning properties of different makes vary considerably. The pad should make sufficient contact with the journal without pressing too strongly, otherwise the oil will be forced out of the pad.

In addition to the worsted spring pad, two small felt pads, fitted in recesses in the side of the keep, are provided to lubricate the face of the

wheel boss; these pads are fed from the oil in the keep by means of holes drilled into the recesses. Easy access to the keep is provided for filling.

For the supply of oil to the brass the mechanical lubricator appears to have advantages over the auxiliary oil box and trimming, since the former delivers a definite quantity of oil for every mile run, whereas the amount of oil siphoned through a trimming feed tends to be erratic due to the difference in the head of oil in the oil box, the viscosity of the oil which is affected by the temperature, etc. The mechanical lubricator consists of a reservoir containing a series of small double-acting pumps, each axlebox receiving its supply from a separate pump unit. One type of lubricator in use gives a normal supply of 2 oz. per 100 miles, but by means of a non-slip thimble any one pump unit can be altered to deliver 3½ oz. per 100 miles; in addition, when the engine is being run in, these amounts can be increased by one-third by means of an adjustment on the driving gear. Regulator valves are fitted to another type so that the delivery of each pump can be adjusted within fine limits to meet all requirements. A back pressure valve, loaded to open at 40 lb. per sq. in., is fitted to each axlebox so that the pipes are kept full of oil when the engine is standing, thus ensuring that the journal receives an immediate supply on restarting.

For coupled wheel axleboxes in which the line of thrust varies considerably in direction in one revolution, it has been difficult to define the best position for the oil to enter the bearing. Until comparatively recent years it was the practice to feed the oil into the crown of the axlebox and spread it by means of an axial groove. When the engine moves from rest, the axlebox is resting on top of the journal and with the groove in this position the oil will not flow out against the pressure. Experiments proved that a better supply was given by feeding through two grooves, each at an angle of 35–40 deg. from the vertical, depending on the diameter of the journal: once in each revolution one source of oil supply is in the region of maximum pressure while at the same time oil from the other feed is supplied at an area of low pressure. Since it was necessary to cut the grooves out of the brass, the bearing metal was split up into three pockets and as it was also felt that the grooves tended to break down the oil film, it is the latest practice on one railway to feed the oil from a mechanical lubricator through a row of holes on the horizontal centre line on each side of the box, thus obtaining an unbroken whitemetal bearing surface to within 1¾ inches of the horizontal centre line (Fig. 2). Another railway has entirely removed the feed to the brass and is experimenting with a felt pad, in place of the worsted spring pad.

For carrying wheels, axleboxes are of gunmetal with a whitemetal

bearing surface giving an arc of contact of 90–95 deg.; oil is supplied solely by means of the worsted pad in the keep.

The journals are turned, ground, and finally lapped with a mixture of emery powder and oil, the latter operation being preferred to rolling.

After boring the boxes, the latest practice is to burnish, an operation which removes the need for bedding-on; when completed, an express engine is given two running trips, light, each of 60 to 100 miles at a

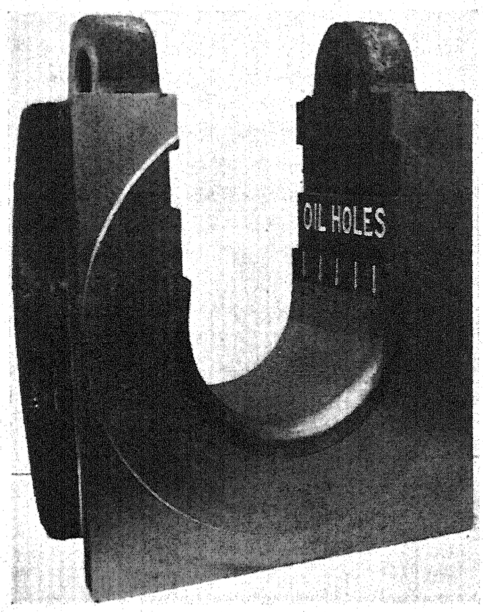


Fig. 2. Coupled Wheel Axlebox showing Oil-delivery Holes and Unbroken Whitemetal Bearing Surface

commencing speed of 25–30 m.p.h. and reaching a final speed of 70 m.p.h., before being put to work.

Owing to improved design of axlebox and journal the bearing pressures due to spring load have been considerably reduced, and now vary between 190 lb. per sq. in. and 250 lb. per sq. in. for coupled wheels, and between 150 lb. per sq. in. and 190 lb. per sq. in. for carrying wheels (calculated on the projected area diameter × length, but neglecting the fillets or radii). For example, the bearing pressure on the coupled axleboxes of a $9\frac{1}{2}$-inch diameter journal

on a modern express locomotive is 193·5 lb. per sq. in. and at a speed of 90 m.p.h. the surface speed at the bearings is 940·5 ft. per min., while the bearing pressure of the $6\frac{1}{2}$-inch diameter bogie journals is 174·2 lb. per sq. in., giving a surface speed of 1,354·5 ft. per min.; all journals are loaded centrally over the full projected area.

The oil used for journals is the same as that applied to the motion generally, as it is not convenient to use different oils for the various parts. The oils used by the four chief railway companies in Great Britain differ in details, but all consist mainly of mineral oil governed by specification as to specific gravity, viscosity, pour point, content of asphaltic matter, and freedom from impurities. Generally, the oil contains a substantial percentage of refined raw rape oil, the amount varying for different classes of work and according to the experience of the company concerned. The following table shows the percentage of rape oil used for the various types of work.

TABLE 1. PERCENTAGE OF RAPE OIL USED IN RAILWAY LUBRICATING OILS

Work		Company			
		A	B	C	D
Express passenger . .	per cent	25	20	15	5
Local passenger and freight	,,	10	10		
Shunting engines . .	,,	—	—		

The specified viscosities of the oils at 140 deg. F. vary, owing to differences in design and work of the locomotives. Two companies specify an oil of high viscosity for use in summer; another company considers that the inconvenience of changing the grade of oil twice a year outweighs any possible advantage, and in its experience no advantage was obtained when the thicker summer oil was used.

Investigations into the causes of hot bearings have included:—

(*a*) Practical examinations and inspection in particular instances.
(*b*) Analysis and examination of oils taken from mechanical lubricators, axlebox keeps, etc., after failure.
(*c*) Statistical analysis of failures having regard to:—

(i) Class of axle bearing and of locomotive.
(ii) Quality and cleanliness of oil.
(iii) Recurrence of failures on a particular bearing or a particular locomotive.

The summarized results indicate that causes of failures are:—

(1) Defects in bearing.
(2) Other defects in locomotive causing abnormal condition in bearing.
(3) Defects in lubricating arrangements.
(4) Presence of grit, water, and inferior oil.

The statistical analysis provided no evidence that the oils normally used by the railway concerned, namely, mineral oil with 20 per cent (for less important work with 10 per cent) of rape oil, were not of sufficiently high quality for their duty; in other words, the evidence was against any assumption that the failures would have been fewer had the normal oils been of higher quality.

THEORY AND EXPERIMENT APPLIED TO JOURNAL BEARING DESIGN

By Professor H. W. Swift, M.A., D.Sc. (Eng.), M.I.Mech.E.*

Theory of Stable Operation

The essential data required from theory and experiment by the designer of journal bearings are the relationships connecting film thickness and friction with the constructional and impressed conditions. It so happens that with partial clearance bearings over the normal working range of eccentricities the discrepancies between the results of different theories so far as these relationships are concerned are quite small. What may in the outcome be a fortunate dispensation for the designer is not so satisfactory to the scientific worker, for he is thrown back on to a more sensitive means of appraisal for theoretical treatments. Such a means is found in the relationship of the impressed conditions to the attitude or angular setting of the journal in the clearance space.

At the present time all theories of journal lubrication suffer because they neglect side leakage. Generally they also neglect changes in viscosity of the lubricant, but it has been found that the use of a representative value for this viscosity does not materially affect the results.

If Osborne Reynolds's basic equation is applied to sleeve or partial bearings with the assumption of a continuous film over the bearing arc it involves the admission of negative local pressures and leads to an unstable form of centre locus for the partial bearing. Moreover, this centre locus, the relationship between attitude and eccentricity, proves to be incorrect in type when subjected to experimental test. If, however, the Reynolds equation is combined with more rational assumptions, it leads to a centre locus in good descriptive agreement with experiment. The assumptions are :—

(1) The pressure rises from zero at the point of "entry", usually defined by the oil distributing groove.

(2) The pressure falls to zero either at or before the point of "departure" of the bearing.

(3) Between the point at which the pressure falls to zero and the point of entry, no material pressure changes occur.

When these assumptions are combined with the condition that the resultant film pressure shall be collinear with the applied load they lead to a continuous range of possible positions of journal equilibrium for any specified conditions of loading. The principles of mechanics

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require that the true journal position is that providing stable equilibrium. With partial bearings of 90 deg. to 180 deg. arc it is found that this position of stable equilibrium corresponds to that particular film of the possible range in which the pressure falls to zero at the same time as the pressure gradient. Stieber* reaches a similar conclusion for sleeve bearings from another rational standpoint.

From the theoretical treatment of any type of bearing the two relationships of direct application in design are:—

$$\frac{P}{\lambda u}\cdot\frac{d^2}{D^2}=\phi_1(\epsilon)$$

$$\mu\frac{D}{d}=\phi_2(\epsilon)$$

and these relationships are most conveniently plotted as curves.

In designing a hypothetical bearing for which these relationships hold, to operate under prescribed conditions of load, speed and lubricant, the clearance d should be chosen so as to provide:—

(1) Maximum film thickness: $t=(1-\epsilon)\frac{d}{2}$.

(2) Minimum friction μ.

From the relationships above it is a simple matter to compute and plot values of $\frac{t}{D}\sqrt{\frac{P}{\lambda u}}$ and $\mu\sqrt{\frac{P}{\lambda u}}$ in terms of the eccentricity. It will be found that the curve for $\frac{t}{D}\sqrt{\frac{P}{\lambda u}}=(1-\epsilon)\phi_1$ reaches a maximum and the curve for $\sqrt{\frac{P}{\lambda u}}=\phi_1^{\frac{1}{2}}\phi_2=\phi_3$ a minimum at certain values of ϵ. The optimum clearance d_o is that which connotes a working eccentricity most nearly corresponding to the best conditions having joint regard to the two curves. Optimum eccentricities for centrally loaded partial bearings are approximately as follows :—

TABLE 1. OPTIMUM ECCENTRICITIES

Bearing arc, deg.	For least friction	For thickest film
90	0·75	0·65
120	0·7	0·5
150	0·6	0·4
180	0·5	0·37

* "Das Schwimmlager", V.D.I.-Verlag, 1933.

But the values of the frictional and thickness criteria do not vary greatly over a fair range of eccentricities on either side of these optima.

Effects of Side Leakage

In applying these principles to the design of journal bearings of normal proportions it is necessary to take account of the effects of side leakage. The analytical treatment of side leakage leads to equations which have only been solved for plane pads. For journal bearings, a certain amount of direct experimental data is available and also a number of results obtained by means of Kingsbury's electrical analogy. A comparison of data from all sources suggests that with any chosen film profile the mean intensity of pressure p, and the frictional coefficient μ, bear certain ratios to their values p_o and μ_o in a film of the same profile but without side leakage, which ratios are not materially dependent on the particular film profile considered.

From this it would appear that the working eccentricity which provides optimum conditions in a bearing without side leakage will also provide optimum conditions in a bearing of ordinary width, so that the values in Table 1 are directly applicable. More recent experiments by Needs on the Kingsbury apparatus suggest that the optimum eccentricity for a 120 deg. clearance bearing becomes greater as the ratio B/D becomes less. But the film profiles in these experiments were selected to conform with Sommerfeld's polar relationship and the differences in operating conditions revealed over the range $0{\cdot}5 \angle \epsilon \angle 0{\cdot}8$ are small. Since, moreover, the choice, *ceteris paribus*, of a smaller eccentricity reduces the freedom for vibration, the values of ϵ_o given in Table 1 may be applied with fair confidence.

For a bearing of axial width B carrying a total load W the dimensionless criteria of film thickness and friction can be written

$$\frac{t}{D^2}\sqrt{\frac{W}{\lambda\omega}}=k_1(1-\epsilon)\sqrt{\phi_1}, \text{ where } k_1{}^2=\frac{p}{p_o}\cdot\frac{B}{D}$$

$$\frac{\mu}{D}\sqrt{\frac{W}{\lambda\omega}}=k_2\phi_3, \text{ where } k_2=k_1\cdot\frac{\mu}{\mu_o}$$

The optimum bearing of a given type will therefore be one which provides for the relevant optimum eccentricity ϵ_o and in which the ratio B/D is so proportioned as to give as high a value of k_1 and as low a value of k_2 as possible.

When values of k_1 and k_2, computed from Kingsbury's results for an eccentricity 0·4, are plotted * in terms of the ratio B/D, it is found

* See Proc. I.Mech.E., 1935, vol. 129, p. 410.

that the thickness factor k_1 falls continuously with the ratio B/D in accordance with expectations, while the friction factor k_2 passes through a minimum value at a certain value of B/D. With 180 deg. bearings this minimum occurs when B/D=3/2 and with 120 deg. bearings when B/D=1 approximately, corresponding in each case to an approximately "square" film in which B/L=1, where L is the peripheral length of the bearing arc.

For a shaft of given diameter D, there will be a certain width B_o necessary to maintain a specified film thickness t_o under a "load" $W/\lambda\omega$. If this width $B_o \leq L$ then a bearing of width L will give optimum frictional conditions with a margin of safety in film thickness. If, however, the width $B_o > L$ then a bearing of width B_o will give the lowest friction consistent with safe film thickness. As D is made less, so the necessary value of B_o increases and with it the difficulty of precision in manufacture and running. Hence it is not desirable to employ a journal of diameter less than the critical diameter D_o which just provides $L=B_o$, and is given by:—

$$D_o^2 = 3{\cdot}1\ t_o\sqrt{\frac{W}{\lambda\omega}} \text{ for a 180 deg. bearing.}$$

$$D_o^2 = 3{\cdot}7\ t_o\sqrt{\frac{W}{\lambda\omega}} \text{ for a 120 deg. bearing.}$$

On the other hand, it is not desirable that the journal diameter should exceed D_o more than may be necessary for reasons of strength and stiffness, since when $D > D_o$ the frictional moment is proportional to D^2.

Working Clearances

Since the capacity of a bearing $\frac{W}{\lambda\omega} \propto \frac{D_o^4}{t_o^2}$ its value is very dependent on the least film thickness t_o and therefore on the least clearance $d_o = 2t_o/(1-\epsilon_o)$ which can be safely employed. In assessing a reasonable value for d_o it is necessary to consider tolerances in manufacture, and the effect of departure from the precise clearance desired. It is found that a variation of 30 per cent from the optimum clearance prejudices the values of friction and film thickness to the extent of 12 per cent and that greater variations have more than proportionate effects. In the light of tolerances permitted in current practice it would appear that for a clearance as small as is consistent with standard workmanship and a variation of ± 30 per cent the value of d_1 is:—

$$d_1=\frac{\sqrt{D_1}}{1{,}000}\left(1+\frac{\sqrt{N}}{20}\right)\geq\frac{3}{2{,}000}\sqrt{D_1}$$

where d_1 and D_1 are in inches and N is in revolutions per minute.

If this clearance is adopted, the critical diameter D_c in inches is given by:—

$$D_c^{\frac{3}{2}}=0{\cdot}012\sqrt{\frac{W}{\lambda\omega}}\left(1+\frac{\sqrt{N}}{20}\right)$$

If we employ the term "specific diameter" D_s to denote the hypothetical diameter of bearing necessary to carry a load of 1 lb. with an oil of representative viscosity 1/2,000 ft.-lb.-sec. units, then:—

$$D_s=1{\cdot}39\left(\frac{1}{\sqrt{N}}+\frac{1}{20}\right)^{\frac{2}{3}} \text{ inches and the critical diameter}$$

$$D_c=D_sW^{\frac{1}{3}} \text{ if } \lambda=1/2{,}000, \text{ or } D_s\left(\frac{W}{2{,}000\lambda}\right)^{\frac{1}{3}} \text{ in general.}$$

The procedure in design is then as follows:—

(1) For given speed N revolutions per minute determine D_s, by formula or curve.

(2) For given load W lb. (and viscosity if abnormal) determine critical diameter $D_c=D_sW^{\frac{1}{3}}$ inches.

(3) For given journal diameter D_1 inches and bearing angle β radians, determine width $B_1=\frac{1}{2}\beta D_1\geq\frac{1}{2}\beta D_c$ inches.

(4) Compute clearance as follows:—

(*a*) If $D_1=D_c$, then $$d_1=\frac{\sqrt{D_1}}{1{,}000}\left(1+\frac{\sqrt{N}}{20}\right)$$

(*b*) If $D_1>D_c$, then $$d_1=\frac{D_1^2}{1{,}500}\sqrt{\frac{N}{W}}$$

Provided the appropriate angle β is employed, this procedure may be applied to 120 deg. as well as 180 deg. bearings. The film thickness and friction will both be somewhat less than with the 180 deg. bearing, but within the range of the clearance formula 4(*a*) the bearing will approximate to its optimum form.

Curves showing specific diameters and clearances will be found in the PROCEEDINGS of the Institution, 1935, vol. 129, pp. 421 and 424. For convenience of designers, line charts drawn up in accordance with the above procedure are shown in Figs. 1 and 2. From Fig. 1 the critical journal diameter D_c for a given load and speed is found by drawing a single line as explained on the Figure. If this diameter is

employed the corresponding clearance is obtained by drawing another line on the same Figure. If the chosen diameter $D_1 > D_c$, then the appropriate clearance is found by drawing two lines on Fig. 2.

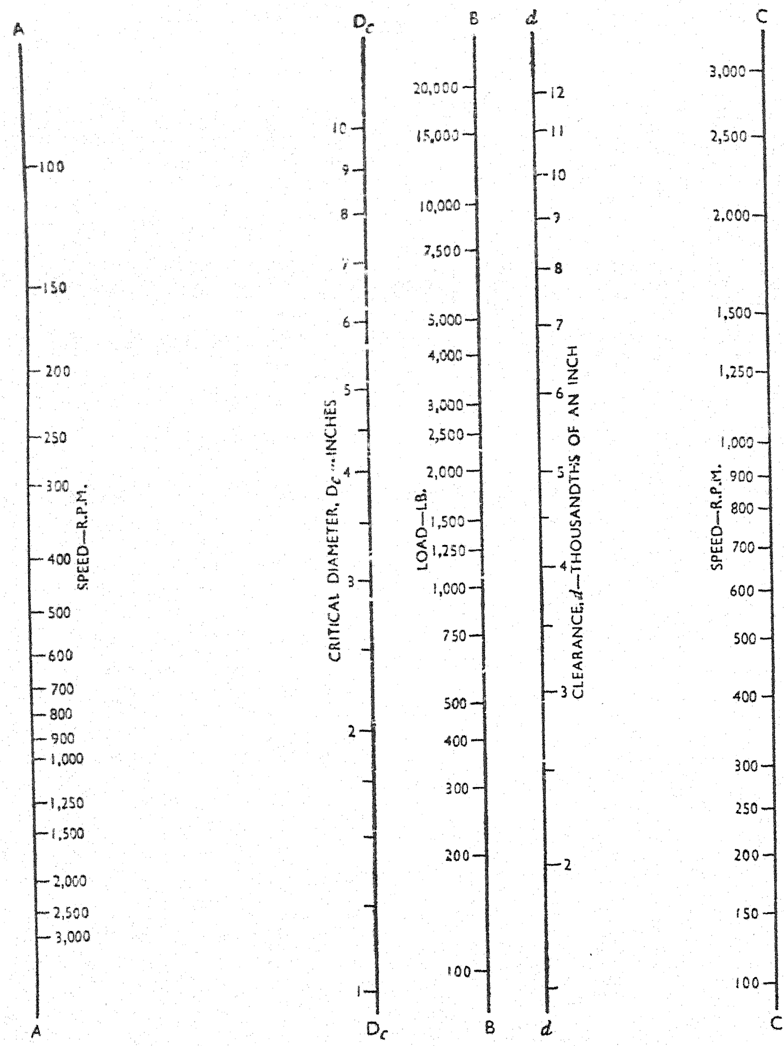

Fig. 1. Critical Diameter and Clearance

Join *speed* on A to *load* on B, cutting D_c in *critical diameter*.
Join *speed* on C to *critical diameter* D_c, cutting d in *clearance*.

FLUCTUATING LOADS

The procedure outlined above is applicable as it stands only to bearings intended to run under steady load. A theoretical examination of certain cases in which the bearing load is subject to cyclic variations suggests, however, that similar methods might ultimately be applied to such bearings as well.

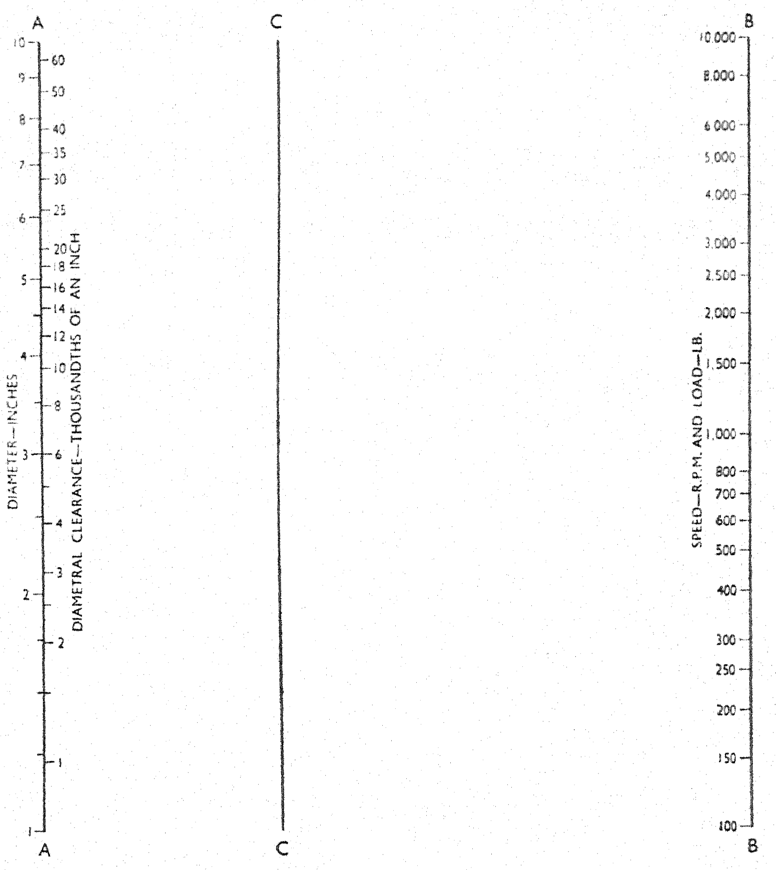

Fig. 2. Optimum Clearance for Journal Bearings, 120–180 deg. Arc

Join *diameter* on A to *speed* on B, cutting C in X.
Join *load* on B to X, cutting A in *diametral clearance*.
The law is: $d_1 = D_1^2/1{,}500\sqrt{N/W}$

With a split sleeve bearing subject to small angular movement and a simple harmonic alternating load, it is found that optimum conditions obtain when the working eccentricity range is about $\pm 0{\cdot}37$, and then the criterion $\frac{p_{max}}{\lambda\omega_1}\cdot\frac{t^2}{R^2}=1{\cdot}48$, where $\frac{\omega_1}{2\pi}$ is the frequency of alternation, as compared with $\frac{p}{\lambda\omega}\cdot\frac{t^2}{R^2}=0{\cdot}61$ for the optimum half-bearing under steady load. At the same time the work per cycle due to reciprocation is $1{\cdot}84\ P_{max}\ t$, as compared with frictional work per revolution of $13{\cdot}8\ Pt$ under steady load.

The case of the bearing for a shaft rotating at speed ω and subject to an alternating load has not been solved for a simple harmonic load, but with a waveform convenient for analysis, it has been found that optimum conditions of film thickness obtain with an eccentricity range $\pm 0{\cdot}4$, and that the criterion of capacity is then $\frac{p_{max}}{\lambda\omega}\cdot\frac{t^2}{R^2}=3\left(\frac{\omega_1}{\omega}\right)^2$ over the range $1 \angle \frac{\omega_1}{\omega} \angle 4$.

The former case has possible significance for crosshead and small-end bearings, while the latter is an approach to the problem of big-end and main engine bearings generally. The frequency of alternations is a factor of primary importance in both cases, and this suggests that the higher harmonics of alternating loads in practice are relatively unimportant. The fact that the maximum load criterion is in each case considerably greater than the load criterion for steady conditions throws an interesting light on the common and successful practice of loading crosshead, big-end, and main engine bearings more severely than bearings under steady load. It is possible that a maximum load criterion of this kind could be employed in the design of bearings for double-acting engines, but in single-acting engines the mean bearing load over a cycle is not small and the conditions under widely fluctuating, as distinct from alternating, loads have not yet been explored.

TESTS OF BEARINGS: METHODS AND RESULTS

By Professor A. Tenot *

The mechanics of friction and the relation of lubrication to friction are assigned a leading place in the curriculum of the French École Nationale des Arts et Métiers. Consequently the physics of lubrication plays an important part in the mechanical laboratory, which includes special apparatus with which the influence of the various factors which enter into bearing design can be studied. This apparatus (Fig. 1)

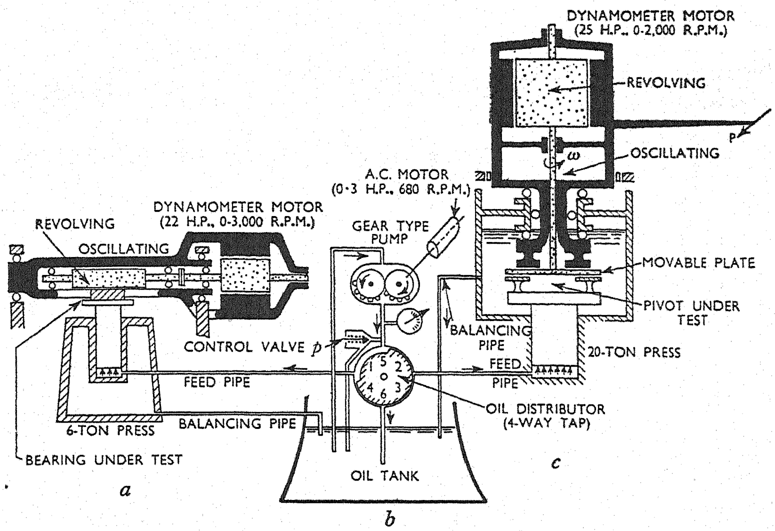

Fig. 1. Friction Testing Apparatus

a Apparatus for testing journal and ball bearings.
b Oil-pressure system.
c Apparatus for testing thrust and footstep bearings.

is in two sections. One is for testing bearings in general, the bearing to be tested being fitted on the piston of a 6-ton hydraulic press, while a variable-speed (up to 2,000 r.p.s.) electrical dynamometer is provided to overcome the frictional torque both of the bearing under test and of the auxiliary ball bearings. This torque is measured by balancing the swinging stator of the motor. Of the six auxiliary ball bearings, two are free to oscillate and four are free to rotate, the latter being mounted inside the swinging frame so that the parasitic frictional torque is eliminated and does not affect the measurements. The

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external frictional torque is balanced out by double weighing. When the motor is stationary, the swinging frame can be keyed to the bearing shaft so that the frictional torque and the coefficient of friction on starting can be determined for various values of load and of the state of the surfaces. The other section of the apparatus, for testing thrust bearings and pivots, comprises a 20-ton press (the bearing under test is fixed on the piston of the press) and a 25 h.p. vertical electrical dynamometer. The press pushes a movable plate between the pivot under test and an auxiliary pivot attached to the swinging frame, thus being inside the system. Parasitic frictional torque is dealt with systematically as before, whilst the remaining parasitic frictional torque is dealt with by the double weighing method. In this way, the coefficient of friction f for each type of pivot or thrust bearing, for each lubricant and each pressure and speed can be determined by measuring the balancing weight Q and the thrust P. Thus

$$C_f = Pfr_{av} = Qd \text{ for } d = 1{\cdot}5 \text{ metres}$$

where r_{av} is the average radius of the thrust bearing and d is the length of the lever arm for the balancing weight Q. The apparatus is provided with an oil pump and a 75 kW. current transformer.

Test Results. By means of the apparatus described the variation of the average coefficient of friction can be plotted against the average pressure p on the bearing surface, the speed of rotation N, the viscosity of the oil $\mu : f = C_f/Pr$. The temperature is kept constant during the tests by water cooling and changes in viscosity in relation to temperature are taken into account where the bearings are not cooled.

For a bearing of known dimensions and clearance (a/r) the coefficient of friction is a function of the three variables μ, N, and p, which can be expressed as $\mu N/p$ (where μ is in kilogramme-seconds per square metre, p is in kilogrammes per square metre, and N is in revolutions per second). The theoretical curve of f as a function of $\mu N/p$ is a parabola of the second degree, as far as the oil film exists: thus $f^2 = k(\mu N/p)$ where k is a dimensionless coefficient. When the threshold of the hydrodynamic state is reached there is a partial or total rupture of the film, leading to boundary lubrication and a rapid increase in friction with risk of seizure. The coefficient f then ceases to follow the parabolic law. This is illustrated by Figs. 2, 3, and 4, relating to smooth bearings of the Michell tilting-pad type. Fig. 2 reproduces the test data obtained with variable velocity and variable pressure on a special bearing with six tilting pads made of cast iron without whitemetal. The journal was made of Martin steel. The tests were carried out without cooling with water or oil. The bearing $(l/d = 0{\cdot}3)$ had a relative clearance

$a/r = 1/500$. The surface of the pad had an area $s = 8 \times 6 \doteq 48$ sq. cm. Up to the critical point $(\mu N/p)$ the f curves are parabolic. With $p_{av} = 111{\cdot}5$ kg. per sq. cm. ($pv = 2{,}300$ kg.-metres per sec. per sq. cm.), $v = 20{\cdot}7$ metres per sec., the values of f and $(\mu N/p)_{crit}$ are very low: $f = 0{\cdot}00187$ and $(\mu N/p)_{crit} = 1{\cdot}5 \times 10^{-8}$ kg.-sec. per sq. metre.

Despite the low value of the relative length l/d these two values confirm the excellent conditions under which tilting pads work. Assuming that almost the entire load (5·35 metric tons) is borne by the lower pad, the maximum deflexion of the free end of that pad

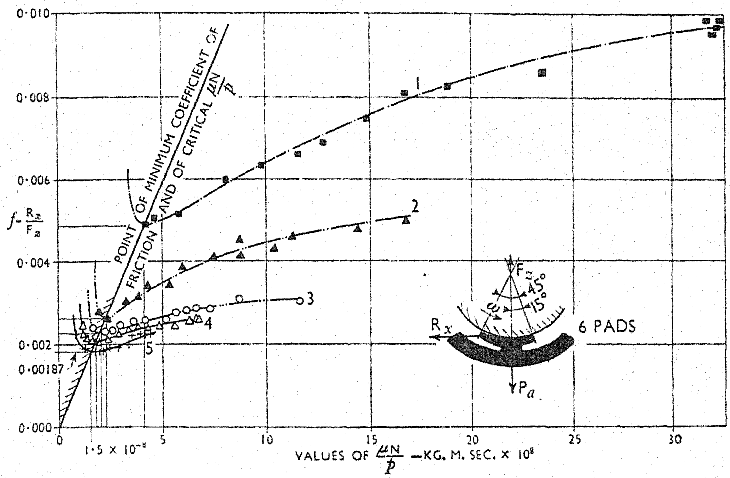

Fig. 2. Tests of a Tilting-Pad Journal Bearing: $f(\mu N/p)$ curves for five values of p

Engler viscosity of oil: 23·7 at 20 deg. C., 3·0 at 61 deg. C., 1·54 at 95 deg. C.

Curve 1=18 kg. per sq. cm.; curve 2=43 kg. per sq. cm.; curve 3 =68 kg. per sq. cm.; curve 4=92 kg. per sq. cm.; curve 5=111·5 kg. per sq. cm. Maximum value of pv attained, 2,300 kg.-m. per sec. per sq. cm.

(assuming that P is 2 cm. out of centre and that E=8,000 kg. per sq. mm.) was calculated to be about 0·07 mm., or roughly of the same order as the maximum film thickness.

For a bearing of known dimensions, theory indicates that all the parabolas $f^2 = k(\mu N/p)$ coincide, and k depends only on the geometrical characteristics of the bearing. Since as many curves are obtained as there are thrusts or pressures, it appears that the pad bends notably.

It will be seen that the curve of the minimum coefficients of friction corresponding to $(\mu N/p)_{crit}$ is a straight line. On the left of the line there is greasy friction and on the right there is fluid friction without

metallic contact. These results are helpful in choosing the characteristics of both bearing and lubricant. To ensure safety, and provide for possible mechanical inaccuracies, these characteristics should not be chosen from near the critical point, though it is not advisable to go too far away from it as f increases with $\mu N/p$, as does the loss of power. The ratio between the value of $\mu N/p$ which is utilized and the value at the critical point can be described as the coefficient of safe stability of the film.

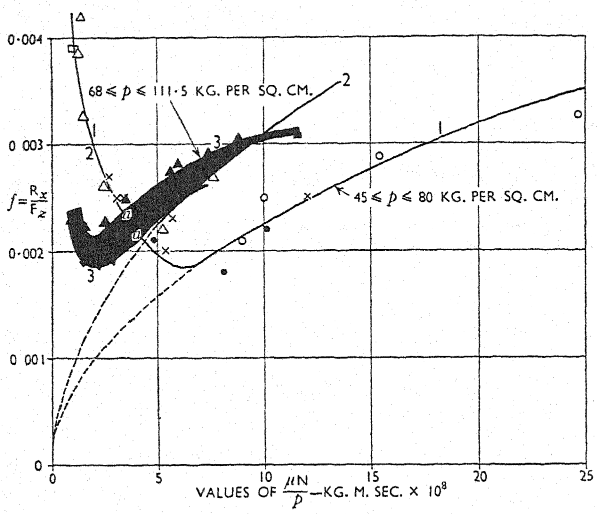

Fig. 3. Comparison between the Author's Results and those of the Isothermos Laboratory

Curves 1 and 2. Isothermos tests on partial bearings at constant speed. The points + =45 are concentrated in the area *aa* where the descending branch (1, 2) of the Isothermos results and the "surface" curve (3) intersect. Bearing arcs in deg.: ○=102, ●=49, + =45, × =41, △=33, □ =24·5.
Curve 3. "Surface" curve. "Châlons-sur-Marne" tests on a tilting-pad bearing at variable speed (up to 20·7 metres per sec.). Measurements indicated at ▲.

Fig. 3 compares the results obtained with a tilting-pad bearing in the author's laboratory and those obtained by the Isothermos Company's laboratory with a bearing with partial brasses lined with whitemetal, the bearing arc being variable from 102 to 24·5 deg.

Fig. 4 gives the results obtained with a tilting-pad footstep bearing made of hardened steel without whitemetal and having a taper of 1 per cent. Here the coefficient of friction includes not only the tangential hydrodynamic reaction of the oil film on the disk but also the friction between oil and disk. The curve of f in relation to $\mu N/p$ is a flattened

parabola, where, on the contrary to the results obtained with the flexible pad bearing, the pressure has but little influence. This indicates that the pivot remains unaffected no matter how great the thrust, and this implies that there is very little deflexion of the pads. With a temperature of 20 deg. C., kept constant by water cooling, it was found that $f_{min}=1{\cdot}6/1{,}000$ and $(\mu N/p)_{crit}=\pm 2{\cdot}5\times 10^{-8}$. This particular bearing has an arrangement by means of which the oil coming from one pad is obliged to circulate and lose heat before it lubricates another pad.

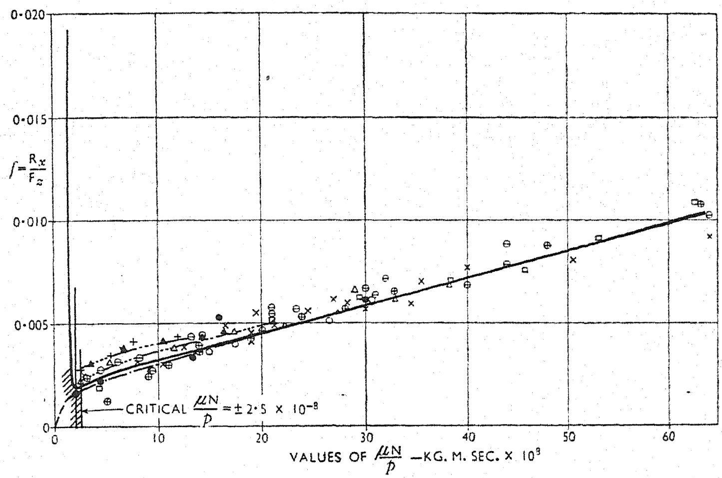

Fig. 4. Tilting-Pad Footstep Bearing (Neyret-Beylier type): $f(\mu N/p)$ Curves for $\theta=20$ deg.

Steel on cast iron; water-cooled; viscosity of oil as in Fig. 1.
Pressures in kg. per sq. cm.: $+=58$, ▲$=46$, ○$=38$, △$=31$, $\times=14$, ●$=7$, ⊖ ⊕$=7<p<38$.

Fig. 5 shows the results obtained with a needle roller bearing of the "Nadella" type. Although this bearing behaved well under test, as average pressures of 120 kg. per sq. cm. were attained at a velocity of 14 metres per sec. without the temperature rising above 80 deg. C., there was no sign of the formation of an oil film (the critical ratio $\mu N/p$ and the parabolic relation $f^2=k(\mu N/p)$ were lacking) when the pressure varied from 60–120 kg. per sq. cm., and $f=0{\cdot}003$–$0{\cdot}004$. Nevertheless the parabolic form of the curve for the coefficient of friction f, for pressures varying from 14 to 43 kg. per sq. cm., and a peripheral velocity v ranging from 0·3 to 14 metres per sec., might be considered as indicating the presence of an oil film between the needle rollers and their support. Rolling and sliding friction would thus

occur simultaneously for the values of p and v in question, though this cannot be stated with certainty.

Conclusions. The work carried out in the author's laboratory proves that French technical education is not unaware of the great part played by lubrication in so many fields. Apart from any question of practical

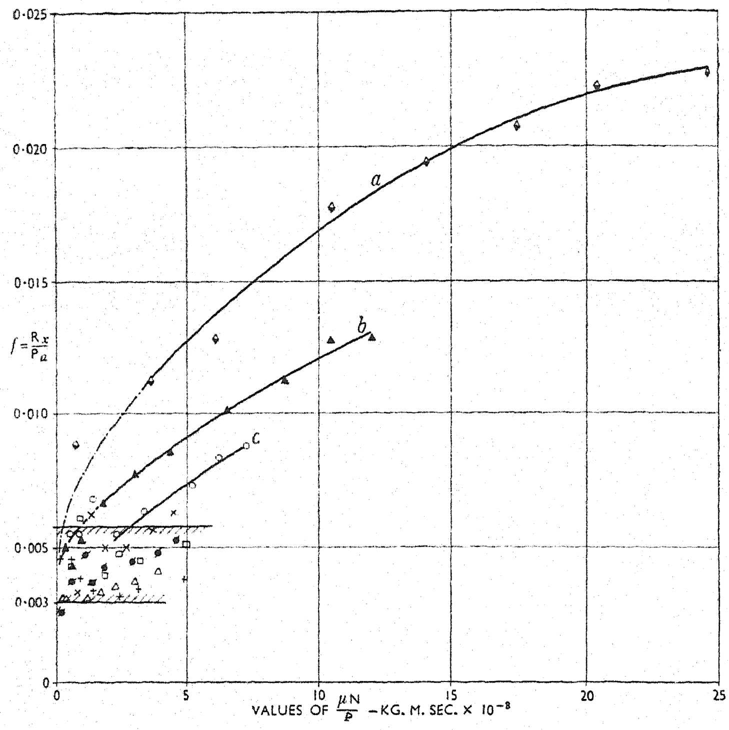

Fig. 5. Tests of a Needle Bearing (Nadella type) under Steady Conditions after Running 3 Hours

Plots for p varying from 60–120 kg. per sq. cm. and θ_{max} from 60–80 deg. Pressures in kg. per sq. cm.: $a=14$, $b=28$, $c=43$.

utility it is claimed that the comparison of experimental results with those obtained by mathematical analysis in a field of fluid mechanics where viscosity plays the essential part, provides a valuable cultural discipline. The accurate finish of the bearings, the influence of the metals employed and their behaviour in use, the preservation of the physical properties of the lubricant, all these constitute subjects which fit well into the general frame of the studies carried out in the French École Nationale des Arts et Métiers.

FILM AND FRICTION MEASUREMENTS ON JOURNAL BEARINGS, INCLUDING FABRIC TYPES

By A. S. T. Thomson, Ph.D., G.I.Mech.E.*

INVESTIGATIONS ON CENTRALLY LOADED CLEARANCE JOURNAL BEARINGS

An investigation has recently been completed on the operating conditions of centrally loaded clearance journal bearings.† The experiments were carried out on a special testing machine fitted with a shaft of 2·5 inches diameter. The arrangement is illustrated in the original paper.† Nine bearings were investigated, comprising three arcs, 180, 90, and 60 deg., and three lengths, 4, 2, and 1 inches, for each arc. Figures in which the experimental values are compared with the theoretical are given in the original paper.† The main results of the investigation may be summarized as follows:—

(1) The results for any one bearing can be represented by single curves drawn on a base of the criterion ZU/P, where Z is the absolute viscosity, U the surface speed, and P the load per unit length.

(2) Increased end leakage raises the value of the coefficient of friction μ. Further the experimental values of μ are in good agreement with the theoretical friction curves calculated from Swift's and Howarth's charts using Kingsbury's side leakage coefficients.

(3) Decrease in length or in arc of embrace lowers the load capacity per unit length. In general the load capacities of the test bearings were lower than those calculated using Kingsbury's leakage coefficients.

(4) The journal displacement diagrams showed a decrease in horizontal displacement with bearing arc or length. The 180 deg. bearings gave very large horizontal displacements. This may be attributed to the dynamic head of oil entering the clearance space. In tests at high values of ZU/P with the 180 deg. bearings the journal centre was observed to rise above that of the brass. The explanation of this feature might be that, due to the large clearance and impact of the oil on the inlet edge, the bearing would run more like an eccentrically loaded one.

(5) The distribution of film pressure over the surface of the nine bearings was investigated for eccentricity ratios ranging from 0·5 to 0·95. The curves obtained showed the change in form of the pressure curve, and the curtailment of the effective bearing arc as the eccentricity

* Royal Technical College, Glasgow.

† Thomson, A. S. T., Proc. I.Mech.E., 1936, vol. 133, p. 413, "Investigations in Film Lubrication".

ratio increased. The pressure determinations were taken by search holes in the bearing from which pipes led to a common header and pressure gauge. An attempt was made to determine pressure variation across the film by means of a search hole in the shaft operating a piston connected to a modified optical engine indicator. The diagrams obtained by this method were flat topped and slightly out of phase. These features resulted from the viscous drag in the search hole, the axial hole in the shaft, and the piston. Also, even the minute movement of the piston necessary to operate the optical indicator resulted in some oil being withdrawn from the oil film. It is thought, however, that if these disadvantages were overcome, say, by using a diaphragm close to the search hole, a rapid and effective method of recording film pressure would be obtained.

In several cases the vertical component of film pressure was equated to the applied load. In general, the sum of the vertical components was about 5 to 10 per cent less than the applied load. Exact agreement could not be expected, as the pressure curves had to be sketched in between the observed points.

Film thickness determinations were made by means of dial gauges bearing on the shaft. These gauges gave little trouble and were definite in action, although possibly not so sensitive as more complicated optical or electrical devices. No measureable film appeared to exist between the gauge plungers and the shaft. Dial gauges were fitted at each end of the bearing, thus enabling any variation in film thickness in a direction at right-angles to that of journal rotation to be noted, and hence allowing truer average values to be obtained. This precaution was justified, as in several cases a considerable variation in film thickness over the bearing length was observed.

The elimination of zero errors in the vertical gauges, due to load and thermal distortion of the bearing housing, are perhaps most simply eliminated by taking zero readings before and after each separate eccentricity determination.

Deflexion of the shaft at heavy loads requires attention, as this deflexion results in unequal thickness of the film in the axial direction. There is also the alteration in clearance due to temperature variations. This effect will become increasingly important as the clearance is decreased. In the present investigation, due to the large radial clearance, namely 0·004 inch, errors from this cause would not be pronounced.

Finally, care must be taken to prevent wear of the experimental bearing as this would further upset conditions. The load, therefore, must be relieved at starting and stopping. Accurate determination of clearance and possible wear will necessitate special measuring instruments.

Investigations on Fabric Bearings

In recent years considerable attention has been given to water-lubricated fabric bearings, notably for roll necks and similar heavy duty. The test bearings were 2 inches long and of 120 deg. arc with a radial clearance of 0·004 inch. In all cases the 120 deg. sectors were cut from the parent 360 deg. cylindrical bush so that the laminations or fabric sheets ran horizontally, that is as shown in Fig. 1. The 360 deg. cylindrical bushes were stoved for 48 hours in light mineral oil. For these experiments the testing machine was modified so as to provide a continuous supply of water to the bath.

From preliminary tests it was evident that the bearings were prone

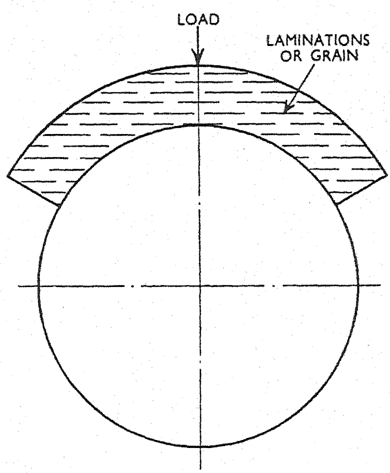

Fig. 1. Direction of Laminations of the Fabric Bearing

to deformation. Thus after running for some time the bearing curvature had increased so that only the edges of the bearing rested on the journal. Under load the bearing would probably spring sufficiently to become a fit on the journal. This would prevent the cooling water reaching the bearing surface and lead to local heating and excessive friction.

As it was thought that this distortion might be due to absorption of water, bearings were immersed in water at various temperatures for several weeks and readings were taken from time to time of the change in form. These tests indicated that the growth due to water absorption was much greater in the direction at right-angles to the laminations than parallel to the grain. Further, the rate of growth increased with

temperature. For example, on the original wall thickness of 0·306 inch the average growth over the bearing surface after 25 days' immersion was 0·0022 inch at 60 deg. F. and 0·0032 inch at 95 deg. F. On continuing the test no further growth was apparent after 25 days. It is obvious from these results that, in bearings having the laminations running as in Fig. 1, the growth due to water absorption will result in a rapid increase in curvature, or decrease in clearance.

The tests were made on a well run-in 2-inch by 2½-inch 120 deg. oil-stoved bearing which, starting with a radial clearance of 0·006 inch, had become bedded to the shaft for about 110 deg. Tests were carried out with water-bath and pressure-water lubrication. In the latter case the water was supplied at 15 lb. per sq. in. to grooves cut near the inlet and outlet edges of the bearing. Two grooves were cut as the bearing was run alternately clockwise and anticlockwise.

In order to simulate the conditions met with in rolling mills fitted with metal bearings, experiments were carried out on a bronze bearing with lubricants ranging from water to heavy oil.

The results and conclusions from the various experiments can be enumerated as follows:—

I. *Water-Lubricated Fabric Bearings.* (*a*) The friction values with water lubrication were rather erratic.

(*b*) No consistent variation in friction with bath temperature was obtained. The reason is thought to be that the fluctuations in friction caused by the non-homogenous nature of the bearing surface were much greater than those which might result from change in viscosity of the water.

(*c*) No definite variation in the coefficient of friction with load was obtained (Fig. 2). This result may follow from the difficulty of maintaining constant conditions due to distortion of the bearing by water absorption.

(*d*) Pressure-water lubrication appeared to increase the value of the coefficient of friction, especially at low rotational speeds. At low speeds with bath lubrication the water was not carried round the shaft and the lubrication was probably due to an oily film on the bearing or journal. This may account for the observed fall in the values of μ at low speeds. With pressure lubrication, however, the μ-speed curve falls continuously from a high static value with increase in speed. A mean curve between the pressure and bath lubrication curves should give a reasonable idea of the probable frictional loss with these bearings.

(*e*) In several cases the coefficient of friction was calculated from the heat given to the cooling water and was found to be in good agreement with the measured value. Conversely, the approximate required rate

of water flow to maintain any desired bath temperaure can be calculated if the load, coefficient of friction, and speed are known.

(*f*) Smearing the journal with oil resulted in a lowering of the coefficient of friction for several minutes, after which it returned to normal.

(*g*) No measurable water film could be detected between the surfaces. This would be expected from the low viscosity of water.

(*h*) A pressure search hole was inserted into the bearing crown. At a load intensity of 1,000 lb. per sq. in. the recorded pressure with bath lubrication was about 2 lb. per sq. in. and with pressure-water lubrication about 15 lb. per sq. in. This result together with the form of

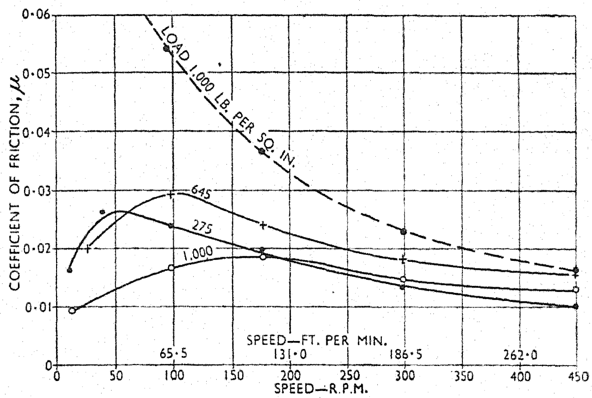

Fig. 2. Friction Characteristics of a 120 deg., 2-×$2\frac{1}{2}$-inch Fabric Bearing Lubricated with Water

——— Water-bath lubrication.
– – – – – Pressure lubrication.

the friction curve and the absence of any measurable film indicate that the lubrication was of a boundary nature.

(*i*) In one test the water supply was cut off. The temperature rose rapidly to boiling point, the friction increased and the bearing became charred.

2. *Comparative Tests between Fabric and Bronze Bearing.* A 2- by $2\frac{1}{2}$-inch 180 deg. bronze bearing with a radial clearance of approximately 0·004 inch was tested with water bath lubrication. At a load intensity of 645 lb. per sq. in. and a speed of 115 ft. per min. the coefficient of friction was 0·175.

(*a*) The measured wear of the fabric bearing for approximately 60 hours' running with water lubrication at loads ranging from 100 to

1,000 lb. per sq. in. was 0·001 inch. The wear of the bronze bearing, water lubricated, for ten minutes' run at a load of 645 lb. per sq. in. was 0·002 inch. It should be noted, however, that the fabric bearing will swell, due to water absorption, and the wear figure given is therefore only an apparent value.

(*b*) The bronze bearing tested at 1,000 lb. per sq. in. with tallow lubrication gave the standard μ-speed curve, with a maximum value for μ of 0·062 at crawl speed and a minimum of 0·002 at the point of viscous film formation.

(*c*) The bronze bearing with tallow lubrication and water circulating through the tallow bath gave erratic friction values varying from those obtained with pure tallow to those given by water.

(*d*) The fabric bearing at 1,000 lb. per sq. in. and with tallow lubri-

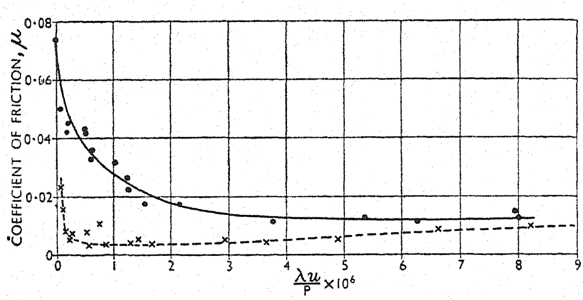

Fig. 3. Comparison of Friction Characteristics for Fabric and Bronze Bearings with Oil Lubrication

——— Fabric bearing, 120 deg., 2×2½ inches.
- - - - - Bronze bearing, 180 deg., 2×2½ inches.

cation gave a maximum value for μ of 0·07 at crawl speed, and values of the order of 0·04 from 150 to 450 r.p.m.

(*e*) The fabric bearing with tallow lubrication and cooling water circulating through the bath gave quite steady friction values of the order of 0·05 at crawl speed, dropping to 0·029 at 450 r.p.m.

(*f*) In Fig. 3 fabric and bronze bearings are compared when lubricated with a heavy mineral oil compounded with 5 per cent tallow to increase its oiliness. The fabric bearing, even when its radial clearance had been increased to 0·004 inch, did not give viscous film conditions at such low values of ZU/P as the bronze bearing.

Summary and Recommendation. From the experiments so far carried out on fabric bearings, oil- or water-lubricated, they would appear to be suitable for heavy duty. In rolling mills fitted with metal bearings

perfect film lubrication conditions will not exist and the lubrication will be generally of a boundary or greasy nature. Under such conditions oil- or water-lubricated fabric bearings will probably give lower frictional loss and considerably longer life. This conclusion is borne out by results obtained in practice. As an example the use of fabric bearings fitted to two welded tube mills resulted in a power saving of 30 per cent. In addition, whereas the previous bronze bearing lasted some eight to ten weeks the life of the fabric bearing appeared to be between eight and nine months.

In addition to the copious supply of cooling water, precautions against rusting and for perfect alignment, it is advisable to provide a large clearance to allow of swelling. One American firm of manufacturers recommends a clearance of not less than 0·3 per cent of the shaft diameter in inches.

The research was carried out under the supervision of Professor W. Kerr, Ph.D., M.I.Mech.E., to whom the author desires to express his indebtedness for much help and advice. He also wishes to express his thanks to the Ioco Rubber and Waterproofing Company, Ltd., Glasgow, who were good enough to supply the test bearings.

TESTS OF PLAIN BEARINGS WITH A NEW METHOD OF LUBRICATION UNDER VERY HIGH PRESSURE

By Professor G. Welter* and W. Brasch†

With the rapid development of machine construction and speedy transport, more consideration should be given to the plain bearing, which still plays an important part in all types of machinery, though it requires technical as well as economic improvement to enable it to compete with ball and roller bearings. Amongst its chief disadvantages, causing loss of power and material, are friction losses in service, particularly on starting and during initial running, and the consequent wear, whilst special attention should be given to its reliability in service and to improvement of its load-carrying capacity. As the frictional resistance, particularly on starting, is abnormally high in present-day plain bearings, it was thought that this drawback could be overcome by a suitable modification of the supply of lubricant, since improvement could not be expected in the bearing materials so much as in the lubrication, which is still in a backward state. Improved lubrication would further be a considerable aid to the bearing metal.

Dry friction or, in favourable cases, half-dry friction, as Falz calls it, causes not only a heavy demand for power when starting, but also wear of bearing and shaft. Further, the use of bearing material which has good running properties but is difficult to run in, becomes possible when fluid friction prevails even on starting.

The improved method of lubrication which is the subject of this paper is designed largely to obviate these difficulties. According to this method, the usual lubrication is supplemented by high-pressure lubrication supplied to the region of highest load at the crown of the bearing. Lubricant is supplied on starting and during service, by means of a pump, in such large quantities and under such a high pressure that, owing to the excess pressure (100–200 atmos.), the oil lifts up the journal from the brass and eliminates the possibility of direct loading. Any metallic contact is thus avoided, as the oil under pressure forms a cushion between brass and journal, so that the frictional resistance in the bearing falls nearly to zero and there will be even less friction on starting than at normal speed.

A high-pressure lubricator, such as is often utilized to lubricate the cylinders of compressors, was used to give the necessary pressure and tests were made on a large bearing testing machine with a 120 mm.

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† Chief Engineer, Frankfurt-am-Main.

journal to find the frictional resistance on starting and during running, with and without high-pressure lubrication. Some results of starting tests at 3–5 deg. C. and at 20 deg. C. with and without high-pressure lubrication are given in the Figures. The frictional conditions of a passenger coach bearing were observed under a load of 6 metric tons, the speed being brought up to 45 km. per hr. (250 shaft r.p.m.), with

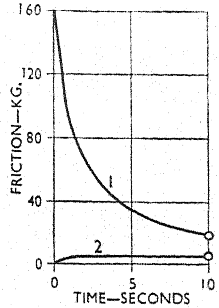

Fig. 1*a*. Portion of Fig. 1 on an Enlarged Scale

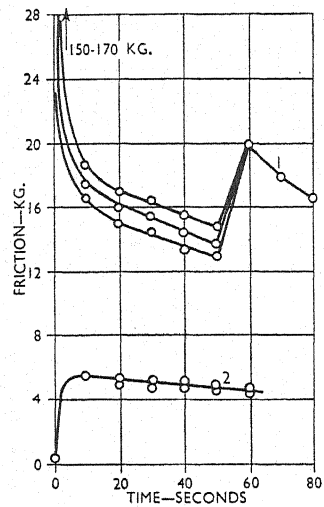

Fig. 1. Friction Observed in 60 Seconds

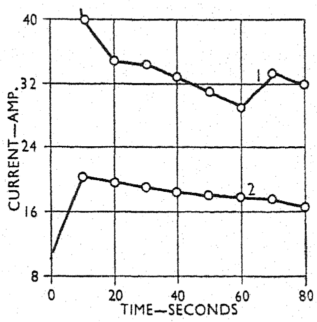

Fig. 2. Current Consumption in 60 Seconds

Curves 1 Without high-pressure lubrication.
Curves 2 With high-pressure lubrication.

and without high-pressure lubrication, within 60 seconds (Fig. 1) and within 10 minutes (Fig. 3) respectively. In Fig. 1*a*, the friction during 10 seconds is shown for clearness on a different scale. The advantage of high-pressure lubrication will be evident. From the extremely high starting resistance (in this case 150–170 kg.) of normal plain bearings, the friction falls comparatively quickly to 17–19 kg. after some 10 seconds, to 14 kg. after 50 seconds more, and, after that, owing

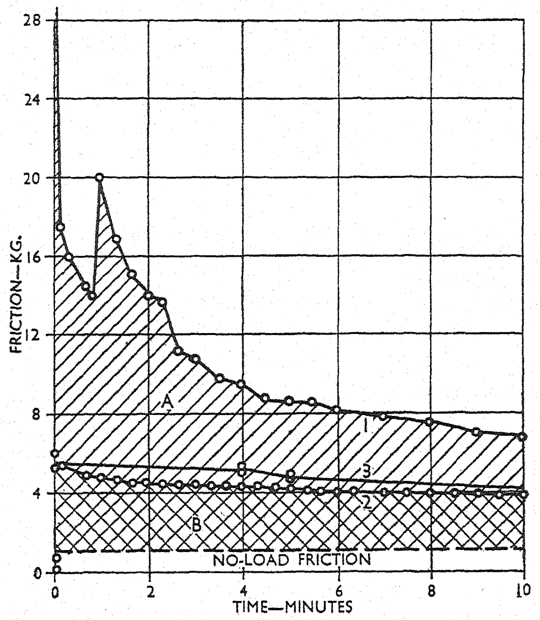

Fig. 3. Friction Observed in 10 Minutes

Curve 1 Journal bearing without high-pressure lubrication.
Curve 2 Journal bearing with high-pressure lubrication.
Curve 3 Roller bearings.

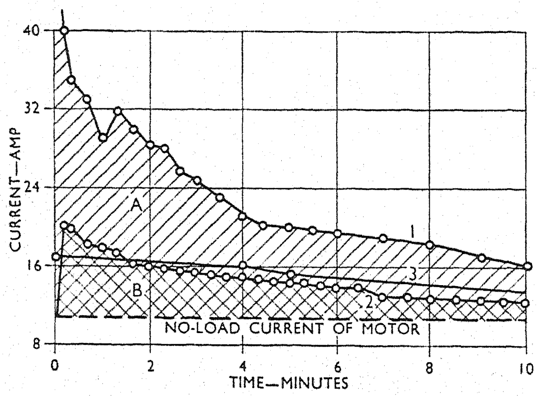

Fig. 4. Current Consumption in 10 Minutes

Curve 1 Plain bearing without high-pressure lubrication.
Curve 2 Plain bearing with high-pressure lubrication.
Curve 3 Roller bearing.

to insufficient oil supply from the pad, increases to about 20 kg. (Fig. 1). It will be seen from Fig. 3 that the frictional resistance of a roller bear-

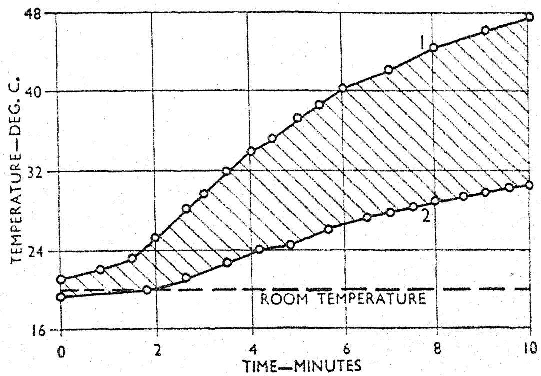

Fig. 5. Temperature Rise

Curve 1 Without high-pressure lubrication.
Curve 2 With high-pressure lubrication.

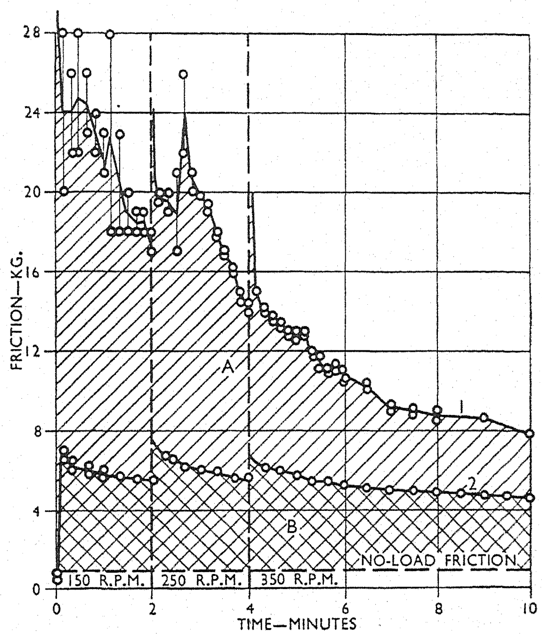

Fig. 6. Friction Observed in 10 Minutes

Curve 1 Without high-pressure lubrication.
Curve 2 With high-pressure lubrication.

ing (3, Fig. 3), tested under similar conditions, is actually higher than that of a journal bearing with high-pressure lubrication. The curve obtained with the high-pressure lubricated bearing takes an opposite course, as it starts near zero (Fig. 1*a*) and rises to a maximum of 5·5 kg. practically without oscillations, gradually approaching asymptotically a minimum of 2·3–2·6 kg. It is noteworthy that under high-pressure lubrication even cold bearings run without oscillating, as the dynamometer shows that the friction takes a steady course (Fig. 1). The results obtained over a longer period—10 minutes—(Figs. 3 and 5) reveal much lower friction and current consumption and a much slower rise of temperature than with a pad-lubricated bearing. The

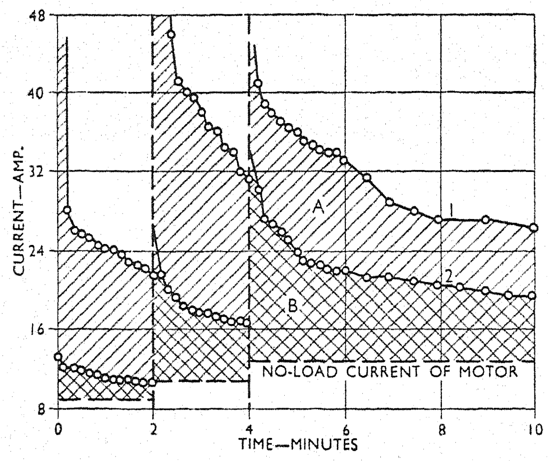

Fig. 7. Current Consumption

Curve 1 Without high-pressure lubrication.
Curve 2 With high-pressure lubrication.

friction found for the journal bearing without high-pressure lubrication (A, Fig. 3) attains a value within the first 10 minutes which is about three times that of the same bearing without pressure lubrication. Similarly with the current consumption of the driving motor (Fig. 4) and the temperature curve (Fig. 5). With a room temperature of 20 deg. C., the temperature increase without pressure lubrication after ten minutes was 27–28 deg. C., whereas with pressure lubrication the temperature increase was only 10–11 deg. C. Still more unfavourable results were obtained with pad lubrication at 2–3 deg. C. (Figs. 6, 7, and 8); the unsteadiness of the friction curve is particularly noticeable, oscillations of up to 10 kg. being recorded in the first 2 minutes (Fig. 6).

The proportion of the frictional surface of the two types of bearing is 1/3 or 1/4 when starting up, sinking later to about 1/2 or 1/3 (Figs. 6 and 7). The temperature difference between bearings with high-pressure and pad lubrication is about 14–15 deg. C. The friction values of the two bearings are actually in the ratio of 1/300 at starting, falling to 1/3 during the first 10 minutes, so that pressure lubrication economizes two-thirds of the power required. Besides this saving in power, pressure lubrication makes it possible not only to economize on bearing materials but to utilize materials that have not yet found such an application. Further, as the temperatures are much lower than those found under ordinary lubrication, highly

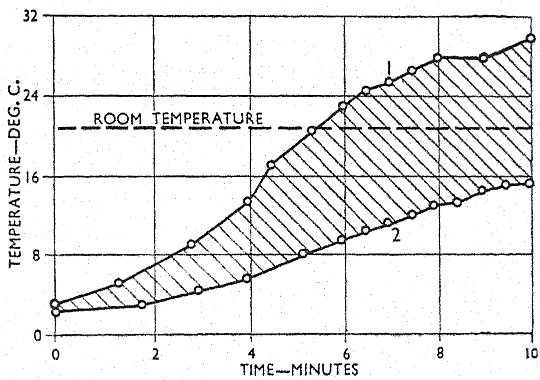

Fig. 8. Temperature Rise in 10 Minutes

Curve 1 Without high-pressure lubrication.
Curve 2 With high-pressure lubrication.

loaded bearings can be reduced in size; lengths can be shortened, thus economizing in bearings and brasses. If highly efficient bearings approached the dimensions of roller bearings a lost field of application could be regained. Should it be desirable to maintain the present dimensions, high-pressure lubrication would make it possible to use materials such as tin-free alloys, which previously were not good enough, for lining the brasses. The supply of oil under pressure greater than the bearing load will also favour the use of such alloys. High-pressure lubrication will also influence the use of aluminium and zinc alloys to line the brasses, as owing to their hardness, these alloys cause difficulties in starting so that hardened journals are required.

TEMPERATURE RISE IN BEARINGS OF AUTOMOBILE ENGINES AND ITS INFLUENCE ON DURABILITY

By C. G. Williams, M.Sc.*

Problems relating to bearing design, lubrication, and materials are causing considerable concern nowadays and there is little doubt that one of the factors most responsible is the steady increase in operating temperatures which modern conditions entail. In the following paper it is proposed to discuss some recent experiments (*a*) on the effect of temperature on the wear and durability of bearings, and (*b*) on the factors affecting temperatures reached in main and big-end bearings.

Bearing Durability. The wear and durability tests were carried out on a machine which was essentially a dummy engine, the bearing under test (0·89 inch in diameter) being inserted in the big end of a connecting rod which was mounted on a built-up crankshaft and, when the machine was rotated at a given speed, the big end was subjected to alternating inertia forces due to the "piston" of known weight. The piston had no crown, so gas compression forces were not introduced. The bearing was maintained at any desired temperature by circulating hot air through the crankcase, the temperature being measured by a thermocouple inserted close to the crankpin surface. The bearing was lubricated under pressure from the crankshaft.

In some wear experiments, a piston giving minimum, maximum, and mean loads of 550, 2,350, and 1,350 lb. per sq. in., respectively, was used, the crankshaft speed being 3,300 r.p.m. and the rubbing speed 756 ft. per min. Crankpins of hardened and tempered nickel-chromium steel and tin-base whitemetal (D.T.D.214) bearings were used. In any particular test a new bearing and a new crankpin were used and, after running-in, the tests were continued and measurements made every 15 to 20 hours for about 100 hours. Experiments were carried out on four oils A, B, C, and D, of high, medium, low, and very low viscosities, respectively.† The mean rates of wear of the crankpin and bearing are shown plotted in Figs. 1 and 2, and it will be observed that, in all cases, wear increased with temperature up to about 130 deg. C. In the region of 100 deg. C. the light oil C gave less wear than the two heavier oils (Fig. 2), though it showed a more rapid

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† For viscosity-temperature characteristics, see Fig. 1 of "Oil Viscosity in Relation to Cylinder Wear", Group II.

increase of wear with temperature (Fig. 1). With the medium viscosity oil B, a temperature increase from 80 deg. C. to 150 deg. C. resulted in

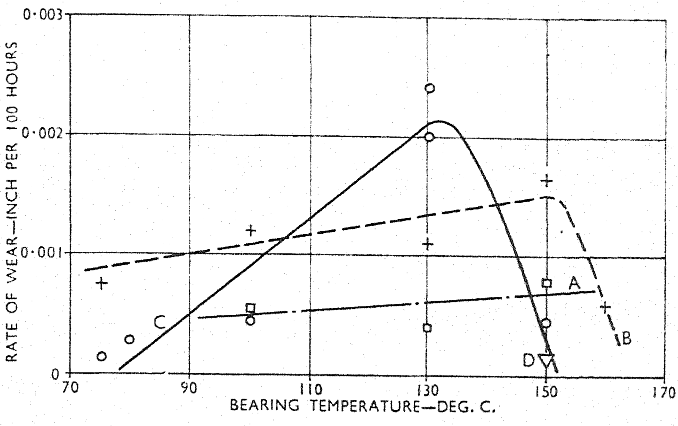

Fig. 1. Crankpin Wear

□ Oil A.
+ Oil B.
○ Oil C.
▽ Oil D.

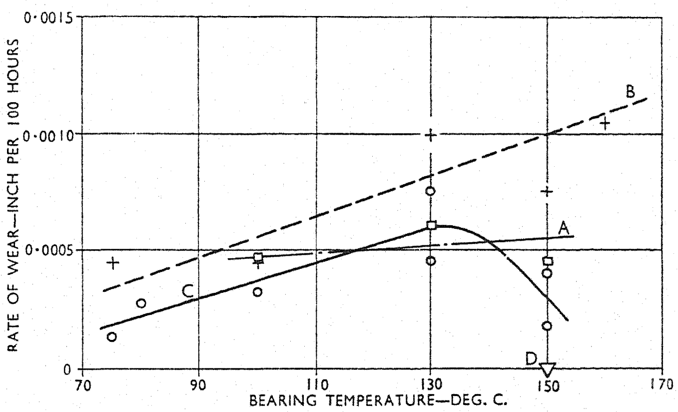

Fig. 2. Bearing Wear

The symbols are as in Fig. 1.

a 70 per cent increase in crankpin wear and a 150 per cent increase in bearing wear. It will be noted, however, that with certain oils, more

particularly the low-viscosity oils C and D, there was a rapid decrease in wear in the region of 150 deg. C., accompanied by the formation of a

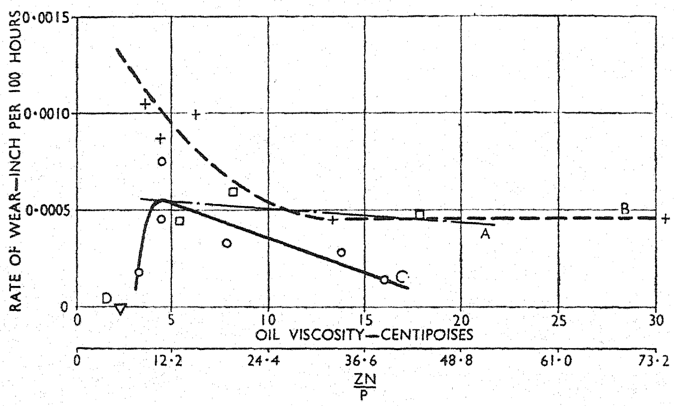

Fig. 3. Bearing Wear

The symbols are as in Fig. 1.

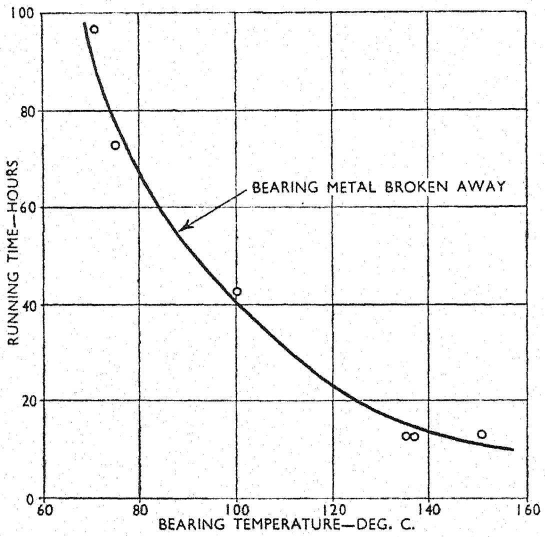

Fig. 4. Effect of Temperature on the Cracking of Whitemetal Bearings

greyish-brown lacquer coating on the bearing surface. Attempts made to cause bearing failure by the use of light oils at high operating tempera-

tures were therefore defeated by the formation of this wear-resisting film. This film, which has been noted by other observers, tends to wear off in subsequent operation at lower temperatures.

The bearing wear figures are plotted against the operating viscosity

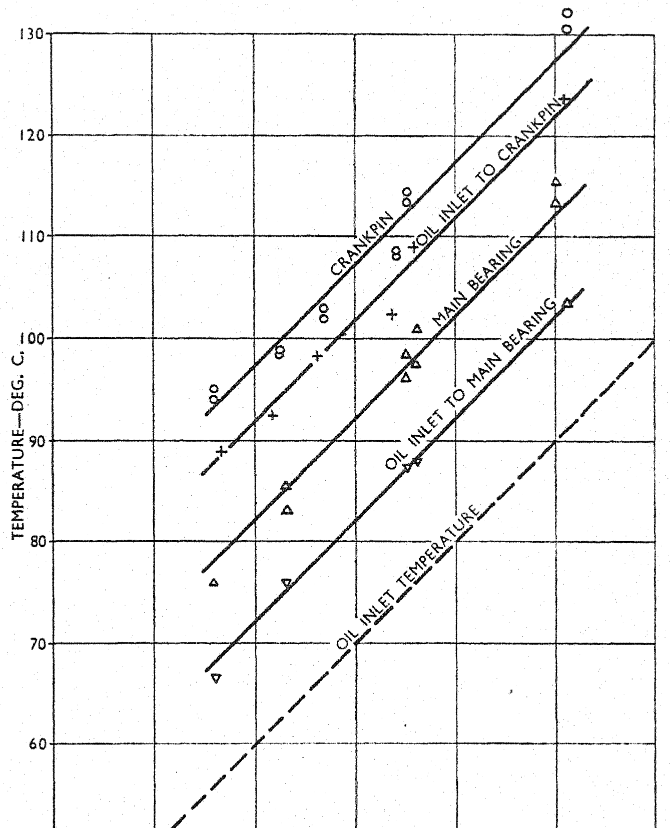

Fig. 5. Effect of Variable Oil Inlet Temperature

Engine speed, 3,004 r.p.m. Oil flow, 37·9 lb. per min. Oil C.

of the lubricant in the bearing, in Fig. 3. The corresponding values of ZN/P (Z=viscosity in centipoises, N=revolutions per minute, P=mean bearing pressure in pounds per square inch) are also indicated. The results suggest that wear is not necessarily a function of ZN/P

since, for example, at ZN/P=35 the wear with the heavier oils A and B was twice that with the light oil C. This, presumably, was due to the fact that, for a given viscosity, the temperature would be lower with the light oil, thereby benefiting the wear resistance of the bearing metal.

There is little doubt, however, that the major bearing trouble experienced within recent years has been the result not of wear but of fatigue cracking, and on the present machine this form of failure was reproduced by increasing the weight of the piston to give a mean pres-

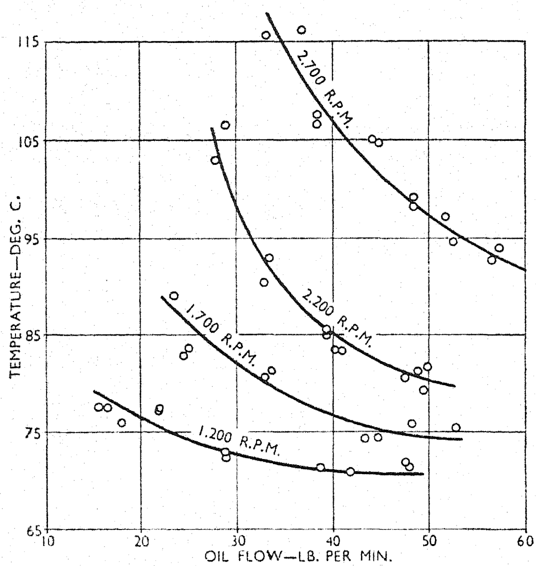

Fig. 6. Bearing Temperatures and Oil Flow

Oil B. Oil inlet temperature, 65 deg. C. Brake mean effective pressure, 40 lb. per sq. in.

sure of 3,080 lb. per sq. in., the minimum and maximum pressures being 550 and 5,850 lb. per sq. in., respectively. By observing at intervals the condition of the bearings it was possible to note the operating time at which pieces of white metal had broken away from the backing. The results of tests at various temperatures (Fig. 4) indicate that bearing failure occurred more rapidly the higher the temperature, e.g. by reducing the bearing temperature from 120 deg. C. to 80 deg. C. the life was increased from 13 to 54 hours, i.e. fourfold, and the form of the graph suggests that, by maintaining the bearing at a sufficiently low temperature, cracking could be postponed indefinitely.

Bearing Temperature Measurements. The bearing temperature measurements were carried out on a six-cylinder 2-litre petrol engine which was converted to a dry-sump engine so that the quantity and temperature of the oil could be controlled. Temperature measurements by means of thermocouples were made: (*a*) near the surface of a crankpin; (*b*) at the oil inlet to that crankpin; (*c*) near the surface of the adjacent main bearing; (*d*) at the oil inlet to that main bearing; and (*e*) at the external oil inlet to the engine.

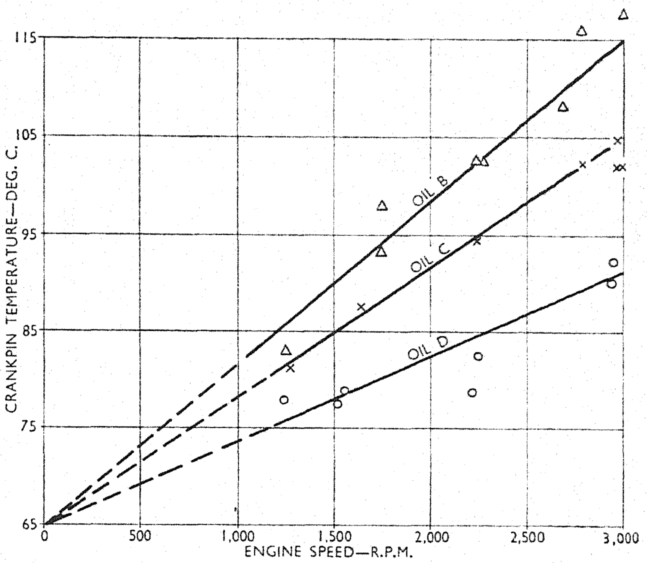

Fig. 7. Speed-Temperature Characteristics

Oil inlet temperature, 65 deg. C. Oil flow, normal.

△ Oil B.
+ Oil C.
○ Oil D.

Measurements made with a variable oil inlet temperature with lubricant C are given in Fig. 5. The most striking feature of Fig. 5 is that the graphs are approximately parallel straight lines, a given decrease or increase in oil inlet temperature having approximately the same effect on all the other oil and bearing temperatures measured. This is a result of considerable importance, particularly in connexion with the efficacy of oil coolers. In addition, these results suggest that the lubricant is responsible for the evacuation of much of the heat generated; and additional evidence for this is contained in Fig. 6,

which shows the effect of rate of oil circulation on the temperature of the crankpin. It will be noted that the effect is considerable, more especially at the higher speeds and, as a first approximation, the temperature rise was inversely proportional to the rate of oil circulation.

From theoretical considerations, the generation of heat in a bearing is independent of the bearing pressure, and experimental results confirming this have been obtained.

Fig. 7 shows crankpin temperatures plotted against speed for the three lubricants, B, C, and D. It will be noted that the crankpin temperatures increased linearly with engine speed, but perhaps the most interesting feature is the important effect of oil viscosity, the lower viscosity oils giving considerably less increase of temperature with speed. The fact that lower temperatures were attained with low-viscosity oils constitutes an important advantage, particularly in regard to bearing life as affected by cracking. In other words, the use of an oil of low viscosity may actually extend the useful life of a bearing.

REPORT ON PAPERS IN GROUP I

JOURNAL AND THRUST BEARINGS

By Professor H. W. Swift, M.A., D.Sc. (Eng.), M.I.Mech.E.

Journal and thrust bearings are conveniently grouped together for the purpose of this discussion because they are at present the only types of bearing which can reasonably claim to take advantage of pressure film lubrication. The advantages of film lubrication are very real, since the frictional losses are a mere fraction of those under any other regime, and continuous running of bearings at high rubbing speeds would be quite impossible without the intervention of a fluid film.

Theory. The science of film lubrication dates from the experimental researches carried out for The Institution of Mechanical Engineers by Beauchamp Tower in 1883, which led to the classic work of Osborne Reynolds published in 1886. Reynolds enunciated the principles of the pressure film, and developed the basic theory of the lubrication of both journal and thrust bearings. Twenty years later, the mathematical theory of the journal bearing was advanced and simplified with the aid of an ingenious parameter by Sommerfeld, and the problem of the plane thrust pad was so completely solved by Michell that his theory forms the quantitative basis of the design of thrust bearings to this day. It is a source of gratification that both these pioneers are among the contributors to this discussion.

In the field of journal bearings, the physical hypotheses of Sommerfeld's theory have required amendment in the light of later researches, including those of Gümbel, Goodman, Brillié, Prandtl, and Thomson; and various assumptions which have been made regarding the effective length of the pressure oil film are discussed by Hanocq. The less rational of these assumptions are gradually being eliminated in the light of measurements of pressure distribution and journal attitude and eccentricity, though a facsimile of Sommerfeld's theory has still many adherents. For most purposes in design, however, the attitude of the journal is a secondary consideration; the more essential relationships are those affecting bearing capacity and friction, and by a fortunate dispensation these relationships prove to be much the same over the working range of eccentricities, whatever theory is followed.

Before the theory of journal bearings can lay any claim to complete-

ness, it must take account of two factors which have hitherto defied exact analysis: the effect of side leakage, and the effect of changing viscosity in the oil film. Some progress has been made with each of these problems. The problem of side leakage has yielded in part to an approximate analysis by Boswall, and in part to an ingenious electrical analogy due to Kingsbury and Needs, which indicates in effect that the known leakage factors for the plane Michell pad can be applied with fair confidence to the journal bearing. The proposal to make use of the established data for plane pads in the design of journal bearings has received encouragement from experiments reported by Prandtl and Hanocq, who have found that the characteristics for bearing brasses of relatively small arc are similar to those for articulated plane pads.

The problem of variation in viscosity due to changes in temperature and pressure in the film, is common to journal and thrust bearings, and does not lend itself to solution by analogy. Nevertheless by making certain simplifying assumptions, Boswall has obtained useful comparative results for the effect of temperature changes in both plane pads and journal bearings, while Bradford and Vandegrift have been led to predict an improvement in bearing capacity, albeit accompanied by an increase in friction, at high pressures. The improvement in friction arising under certain conditions from the use of fatty oils has been explained in terms of their relatively flat pressure-viscosity relationships. Needs supports this view, but points out that such oils also show to advantage under static conditions.

Experiment. On the experimental side of the problem of bearing lubrication, the intrinsic difficulties of technique have taxed the ingenuity and skill of many workers, and much of the earlier work must now be regarded as exploratory. But, following an appreciation of the importance of infinitesimals and an improvement in the precision of manufacture and measurement, progress has been rapid, and the capacity and friction characteristics of bearings under steady loads can now be explored with confidence. Interesting and varied examples of recent experimental technique will be found in the contributions by Prandtl, Hanocq, Odqvist, Linn and Sheppard, Clayton, Jakeman and Fogg, and Thomson.

In the past, the chief purpose of systematic experiment has been to ascertain to what extent theory is capable of predicting the performance of a bearing, and to provide interim data useful to the designer. The experimental results embodied in the contributions under review are evidence of the progress made. Experiments reported by Thomson confirm that under film conditions the significance of load, speed, and lubricant can be jointly expressed in the dimensionless product $p/\lambda\omega$.

Prandtl shows that the extent of the pressure film depends on mechanical stability rather than the length of available bearing arc, so that excessive bearing arc increases the frictional losses of a bearing without increasing its capacity. Kingsbury's analogy indicates that optimum frictional conditions are obtained with a bearing surface of approximately square proportions, and Boswall finds that offset as compared with central loads tend to improve the capacity and reduce the friction of a bearing. Finally, Thomson has found that film thickness and friction in a bearing can now be predicted with reasonable confidence by adopting Kingsbury factors in conjunction with theoretical results. In short, theory and experiment have now been brought into sufficiently good agreement to inspire fair confidence in each, and they have made possible the enunciation of rational procedure in the design of partial bearings under steady loads, giving results which Haslegrave* has found to be in reasonable accord with the more advanced current practice.

Having perfected his technique and effected a satisfactory liaison with general theory, the experimental worker is now tending to turn his attention to more special problems which are of direct interest to the designer, but are not at present amenable to theoretical treatment. At the present time particular attention is being devoted to the conditions of breakdown of the pressure film, to the unsteady conditions which arise from vibration or cyclic fluctuations of load and, in co-operation with the designer, to the development of new bearing materials and lubricants.

Investigations of the conditions of seizure in journal bearings, notably by Clayton, Jakeman and Fogg, Hanocq, and Tenot, give general support to the descriptive theory of film breakdown expounded by Heidebroek. It appears that at some value of the load criterion, depending on the truth and finish of the bearing surfaces, contact becomes imminent at the point of nearest approach, somewhat in advance of the load line. The more perfect the surfaces, the greater is the critical value of the load criterion, the nearer the point of contact to the load line, and the lower the corresponding coefficient of friction. At the critical point, metallic contact and wear and the consequent rise in friction can be alleviated by "oiliness" in the lubricant. Deficiency in "oiliness" or excessive load causes sharp increases in friction and wear. It is clearly established that the load criterion itself does not determine either the incidence of "boundary" conditions, the minimum friction, or the rate of subsequent wear. But, as Clayton points out, a wide field of investigation is still open, and information is particularly needed regarding the effect of wear and bedding on the capacity of a bearing.

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* Proc. I.Mech.E. 1935, vol. 129, p. 435.

Turning to the other extreme, another matter of concern presents itself. At small values of the load criterion, evidence is given by Guy and Smith, Newkirk, and Soderberg of systematic vibrations which gradually disappear as the load is increased. This form of vibration is well known to manufacturers of turbo-generating machinery, which works at high speeds and relatively light loads. It is attributed in some cases to slight want of balance, and in others to film conditions tending to perpetuate the natural vibrations of the rotor, but its nature is by no means properly understood, and Guy and Smith emphasize the need for its systematic investigation. Since the pressure film in a journal bearing is theoretically incapable of withstanding a load alternating at half the journal speed, there is perhaps significance in the experience of Soderberg and Newkirk, that vibration troubles tend to commence at twice the critical speed of the rotor.

Design. Attention may now be turned to the bearing designer, whose needs, after all, should be the pivotal consideration in any technical study of lubrication. In a general way the designer realizes that the more nearly he can apply the principles of hydrodynamic theory and approach the optimum conditions of film lubrication, the closer will he be to perfection. He is familiar with the significance of the load criterion, he appreciates the prime importance of film convergency in the appropriate region, and he realizes that, from the hydrodynamic standpoint, there are optimum proportions, bearing arcs, load lines, and clearances.

But in the main, the designer has not been able to rely on theory and experiment; he has been compelled to work in the light of his own experience, often by trial and error, and with only such qualitative assistance from theory and experiment as he could apply to his special needs. He has, moreover, been brought to realize that, even when the theory of lubrication has been worked out to finality, the design of a bearing can seldom be based purely on considerations of minimum friction and optimum film thickness, and that many conditions must arise in practice where the maintenance of any kind of pressure film is technically impossible.

He realizes with McKee that the working temperature of a bearing is often a primary consideration, not only by reason of its effect on the viscosity of the lubricant and so on the working eccentricity and load capacity, but also on account of the distortion which temperature differences may produce in the geometry of the film. For this reason he is compelled to consider the cooling effect of the lubricant, and to face the problem of heat transmission through the shaft and bearing. In order to ensure a safe working temperature, the designer may need, as

Soderberg points out, to employ for cooling purposes far more oil than is required to maintain the lubricating film, and he may need to design his bearing to provide means of free oil circulation, even at the expense of load capacity. Since Lasche's experiments thirty years ago, little systematic information has been made available regarding heat flow and dissipation from bearings, and as running speeds have become higher this matter has become increasingly important. McKee suggests an approximate expression for heat dissipation in terms of temperature rise, which differs, as it must, according as the bearing is self-contained or independently cooled, while Soderberg puts forward an empirical formula suitable for turbine bearings.

The supply of oil necessary for the pressure film of a bearing of known dimensions can now be estimated with fair accuracy by means of the Karelitz formula expounded by Juhlin and Poole, though it is very dependent on the disposition and extent of the distributing groove as well as the bearing clearance and speed. It is usual, of course, to allow a considerable margin of safety on this estimate, apart from any cooling supply which may be required.

The mechanism of oil supply by means of oil rings and collars has been successfully investigated by Karelitz and Baildon, who have independently drawn attention to the advantage of rings with internal circumferential grooves for high-speed work, the former referring to Baudry and Tichvinsky's work. Baildon also offers guidance in the dimensioning of oil rings, and shows that radial depth or axial width is a leading consideration according as the shaft speed is low or high.

When mechanical or so-called forced lubrication is employed, the distribution of oil to the different bearings of a single supply system is apt to be uncertain if it depends on clearances and oil grooves alone; it is more effectively controlled by means of nozzles in the case of turbine machinery or by separate metering pumps in the case of locomotives.

True forced lubrication under pressures from 1,000 to 3,000 lb. per sq. in. is applied to certain important bearings to maintain an oil film at and near standstill, and continuous lubrication of this type, which has been introduced on electric coaches in Poland, is said by Welter and Brasch to lead to economy in the size and material of bearings.

Another consideration which compels the bearing designer to deviate from theory is the impossibility of attaining geometrical perfection under practical conditions. As Falz points out, the proportions of bearings and their rated capacity have to be determined according to the precision and surface finish obtainable in manufacture, and with due regard to the effects of elastic bending, tilting, and imperfect alignment on the distribution of load and the geometry of the film.

The truer the axes of the journal and bearing under load, the more uniform the film and pressure distribution. The more perfect the bearing surfaces, the thinner the film which can be tolerated with safety and the greater the effective capacity of the bearing. The narrower the bush, the less will be the effect of bending and tilting, but, on the other hand, the greater the leakage of oil and the lower the basic capacity of the bearing. So-called self-aligning bearings modify the effects of tilting, but are no remedy against bending or misalignment of a series of bearings. Self-aligning devices which operate by sliding are criticized by Falz as uncertain in action, while those which depend on rolling contact involve high inter-metallic pressures and impair the conduction of frictional heat.

Bearing Materials. The choice of bearing material and lubricant is another matter on which hydrodynamic theory gives little guidance. Insofar as he is able to ensure operation within the regime of the pressure film, the designer is not greatly concerned with any physical property of the lubricant except its viscosity (for which he can compensate in his design), and his chief concern with a bearing material is that it should withstand the necessary load without deformation, and be capable of accurate manufacture and high finish. But in the majority of cases, where bearings require at least to start under "boundary" conditions, the designer is glad to take some advantage of "oiliness" in a lubricant, and he has to give more careful consideration to the properties of bearing materials.

The ideal bearing material needs a combination of almost incompatible properties. It needs good mechanical strength over the working range of temperatures and sufficient structural stiffness to prevent general distortion, and yet sufficient plasticity to bed itself without wear. It must be capable of high surface finish, and yet be free from abrasive action either in the mass or in its fine products of wear. It must have a low intrinsic coefficient of friction and high thermal conductivity. No such material has, of course, been found, but a good deal of progress has been made in recent years towards the development of properties most desirable for specific applications, and some interesting results have been submitted for discussion.

In engineering practice, bronze bearings have been generally favoured for heavy pressures and slow speeds, particularly where alignment and lubrication are good, while whitemetals have been used with higher speeds or less perfect alignment. In the range of copper alloys, Neave and Sallitt are able to report some interesting developments. They find that the wearing capacity of bronze bearings can be improved by increasing the tin content as far as 20 per cent. For higher

speeds and less perfect conditions, plastic bronzes have been developed containing up to 25 per cent of lead, and for such heavy duties as are encountered in automobile crankshaft bearings, a thin copper-lead lining is claimed to withstand higher loads and temperatures than tin-base bearing metals. For medium duties, porous copper-tin bearings moulded from powder lend themselves to multiple production, and are able to conserve oil up to 30 per cent of their volume.

The specially severe loading on the bearings of certain internal combustion engines has led in some cases to cracking trouble with tin-base bearing metals. This has been diagnosed by Macnaughtan as fatigue due in the main to thermal stresses, and he suggests that the fatigue resistance of these metals over the useful range of temperatures can be improved by freedom from lead and by the addition of cadmium up to 2 per cent. The fact that cracking can be largely overcome by keeping the bearing temperature low has led Williams to the interesting suggestion that bearing life may be extended by the use of oils of low viscosity.

Concurrently with this work on more or less traditional bearing metals, important developments have taken place in non-metallic bearing materials, particularly synthetic resins and vulcanized rubber, and in each case mainly with water lubrication. Several contributors, including Beuerlein, Eyssen, Rochester, Rowell, and Watson Smyth, are impressed by the potentialities of fabric bearings. For severe duties, such as rolling mill bearings, synthetic resin is being employed, built up with fabric into laminated form, while for less onerous duties it is often moulded with felted fibres and sometimes with graphite. The laminated type involves certain obvious limitations in design, but, on the other hand, its capacity is rated as high as bronze or Babbitt metal, and in rolling-mill applications it has been found superior as regards friction, durability, and general performance.

On account of their tendency to flow, all plastics require adequate backing and framing by the bearing shell, and on account of their low thermal conductivity they require an adequate and continuous supply of water for cooling. Grease is sometimes injected to prevent corrosion during standstill and to facilitate starting, but this has been found unnecessary if a closed system of circulation is adopted in conjunction with soluble oil. The swelling effects of water and oil have received attention from Beuerlein and Thomson, and special rules have been formulated on the Continent to make allowance for these effects in the design of fabric bearings.

Rubber-lined bearings are specially suited to water-exposed conditions which have in the past provided the traditional field for lignum vitæ. Brazier and Holland-Bowyer have found that, with water as combined lubricant and coolant, a suitably designed rubber bearing

compares favourably with metal bearings in the matter of friction, and can withstand high pressure at moderate speeds. As in the case of fabric bearings, it is important to ensure adequate water circulation; and for this reason, when water is supplied under pressure, helical grooves are often provided on the bearing surface. For general purposes, a fluted or polygonal bearing surface is favoured because of the freedom for circulation it allows, although tests by Fogg and Hunwicks at the National Physical Laboratory show that a plain cylindrical rubber bush with good clearance produces considerably less frictional resistance.

Thrust Bearings. If there is one type of bearing which owes more than any other to analytical theory, it is the pivoted-pad thrust bearing. Designers, represented in this instance by Dowson, Gibson, and Kraft, are fully conscious of their indebtedness to Michell and those who later elaborated his theory, and it is significant that the proportions and pivot location shown in Michell's original patent specification are still standard practice. Approximately square pads are pivoted about four-tenths of their length from the forward edge by some form of point or line support according to the thrust to be transmitted. The pads are usually not less than ten in number, and are of steel or bronze, with whitemetal linings. Pressures up to 450 lb. per sq. in. and mean rubbing speeds up to 170 ft. per sec. are commonly employed. Except for vertical-shaft applications where, as Gibson points out, the full thrust is effective from standstill, these figures are only limited by the means for heat removal. In a thrust bearing of normal design the pivotal construction is not favourable to heat conduction from pads to housing; consequently, greater reliance has to be placed on the oil for cooling purposes than in a journal bearing, and it is all the more necessary, as emphasized by Dowson and Soderberg, to determine the oil circulation on the basis of heat disposal.

The problem of equal partition of the load between the several pads of a thrust bearing does not now present serious difficulty; some form of elastic backing or automatic compensation is sometimes incorporated, but, in the main, reliance is placed on accurate multiple production.

Accuracy in machining has also contributed to the success of the tapered-land thrust bearing, described by Linn and Sheppard, and used by one large American firm for steam turbines of all sizes. Appropriate tapers for the lands in the radial and peripheral directions are determined from theoretical principles, and tests show that the taper should be different at the outer and inner circumference. These bearings carry ratings of the same order as the pivoted pad type.

Michell, Gibson, and Odqvist show that the pivoted pad principle can be applied with success to journal bearings, resulting in a bearing

which is economical in space and independent of the direction of loading. The capacity of these bearings is determined largely by considerations of heat disposal, and is otherwise only limited by the rigidity of the component parts and the strength of the pivots. Pivots with spherical seatings have been proposed by Michell, with the purpose of improving the conditions of heat transmission, and at the same time increasing the strength of the bearing. When a bearing of this type is required to operate in either direction of rotation, it is desirable to change the position of the pivotal axis. In the Nomy bearing described by Odqvist, this is effected by causing the blocks to rotate with the shaft, and mounting them so that they can tilt to form a convergent film in either direction.

Returning now to the more general aspect of film lubrication, the extent to which the designer is able to rely on theory and experimental research, and the extent to which he has to depend on practical experience and service tests may well be illustrated from contributions which deal with practice in three widely different bearing applications: turbine machinery, internal combustion engines, and locomotive axles.

Turbo-Generator Bearings. Guy and Smith have compiled a comprehensive statement of the practice of the principal steam-turbine manufacturers in this country, and Soderberg shows that a similar practice is followed in America. Owing to their high speeds and moderate pressures, turbo-generator bearings afford specially good opportunities for the application of hydrodynamic principles; and general practice employs bearing arcs up to 120 deg., ratios of axial width to diameter of about $1\frac{1}{2}$, clearance ratios of 2/1,000 or more, and frequently a substantial relief in the upper half of the bearing. The main problem in turbo-generator bearings is heat disposal rather than power loss. The adopted relationships between heat loss and temperature rise are purely empirical, and the formulæ for computing heat loss are themselves also partly empirical, since they need to take account of losses in the idle bearing arc, and also, in the case of turbines, of heat conduction from the steam. The actual oil flow is normally much in excess of that theoretically required, a relatively small proportion passing through the film proper, while the remainder is circulated in a systematic way for the purpose of heat removal. In this connexion, interesting developments may result from an encouraging series of tests on a shop turbine reported by Samuelson, in which water containing 2 per cent of soluble oil was successfully used as a lubricant. This is of special interest in connexion with the fire hazard, which is discussed by Chittenden and by Barclay. The problem of oil film vibration, which, as already mentioned, is of special concern to designers

of turbine bearings, has sometimes been solved by reducing the bearing arc or introducing a self-imposed load, as in the Newkirk bearing.

Internal Combustion Engine Bearings. In comparison with turbo-generator bearings, Taylor is of opinion that the main crankpin bearings of internal combustion engines can rely very little on hydrodynamic theory at the present time. Their design is therefore largely based on experience and trial, and there is a marked reticence in some quarters to disclose the methods employed. The empirical character of these methods is ascribed by Taylor partly to want of rigidity in the bearings and uncertainty of the relationship between heat dissipation and temperature rise, but more particularly to the cyclic fluctuations of the bearing load which, incidentally, prove surprisingly beneficial to bearing capacity.

Mickelsen gives evidence that in high-speed engines the inertia forces predominate over those due to combustion, to such an extent as to form the better basis for design. In order to maintain a pressure film under the very heavy loads encountered in these bearings, full advantage is taken of the available bearing surface, and oil grooves are avoided as far as possible. Dicksee has emphasized the disadvantage of the ordinary system of forced lubrication from this standpoint, and also because of the unnecessary circulation of dirt which it involves.

Railway Axleboxes. Railway axle bearings, although they inspired some of the earliest research work on film lubrication, have not, according to Petree, profited greatly from it in the past. Difficulty in achieving pressure film conditions has arisen partly from the necessity for starting with full load on the crown of the bearing, and partly from the practical inconvenience of ensuring a profuse supply of oil. Stanier points out that the horizontal component of load on the coupled axles gives them some advantage over the axles of carrying wheels as regards starting conditions, but its variable character complicates the problem of locating the distributing grooves. In more advanced practice, locomotive bearings are supplied from a mechanical lubricator or auxiliary oil box in addition to the worsted spring pad, and temperature measurements reported by Stanier suggest that film lubrication is probably attained when running at speed.

The running qualities of the more important bearings have been improved by keeping bearing pressures below 200 lb. per sq. in. and by reducing the bedded arc to 90 deg., and Petree mentions laboratory tests which have been made with multiple brasses to accommodate variations in the direction of load. Self-contained devices to improve starting and running conditions by ensuring immediate and profuse oil

supply, appear to be more popular on the Continent than in this country. The "Isothermos" axlebox employs disk or palette lubrication with overhead supply channels, while the Peyinghaus bearing, described by Petree, relies on an under-brass, clear of the journal but close enough to retain an oil reserve when standing.

The three types of bearing chosen as examples support a feeling which is borne out generally by the contributions to this group, that no bearing application is so simple that it can rely entirely on theory or so intractable that theory can afford no help.

There are many fields open for investigation, both theoretical and experimental, and many unsolved problems to which attention has been drawn by contributors. Some of these are within the scope of the ordinary laboratory, while others require close collaboration between the designer and investigator, and a few, notably those relating to turbine bearings, can only be successfully undertaken by extensive co-operative research. While so many problems remain unsolved, and so long as technical progress in engine and machine design is limited by bearing considerations, there is no room for complacency, but there is some comfort in being able to conclude with the thought that the designer on his part has become conscious of the value of research, while theorists and experimenters appreciate the complexity of the problem confronting the designer and realize that there are more things in heaven and earth than are dreamt of in their philosophy.

DISCUSSION ON PAPERS IN GROUP I

13th October 1937

The General Discussion on Lubrication and Lubricants was opened by Sir H. Nigel Gresley, C.B.E., D.Sc., M.I.Mech.E. (*Past-President*), at the Central Hall, Westminster, on Wednesday, 13th October 1937, at 2.30 p.m.

The CHAIRMAN, in opening the meeting, said that, to his very great regret, he had to announce that the President, Sir John E. Thornycroft, K.B.E., was unable to be present owing to illness, and on his behalf, it was first his (the Chairman's) very great pleasure, as Past-President of the Institution, to extend a very hearty welcome to all present. It was a matter of some regret to the Council and himself that the meetings could not be held in the Institution's building, but the large attendance made that impossible.

They had met to discuss a subject of vital importance to engineers, one to which the Institution had always devoted considerable attention. In fact, the Institution could reasonably claim with some pride to have played a leading part in the foundation of its experimental and mathematical study in England.

The Discussion was organized because it was recognized that, although lubrication was receiving extensive study and still presented many unsolved theoretical and practical problems, yet there existed, scattered throughout the world, a vast amount of knowledge and experience, the correlation of which had never previously been systematically attempted. It was felt that the time was ripe for such an attempt to be made, and that a most useful method to adopt would be the arrangement of a general discussion to be based on a number of papers to be invited from a widespread and authoritative authorship. The Council of the Institution of Mechanical Engineers decided to assume the responsibility for arranging such a discussion. But it was recognized that the problem concerned every branch of engineering and every nation, and therefore the collaboration was invited of certain other technical Institutions and Societies both in Great Britain and abroad. The response to that invitation had been such that no fewer than 31 British and 22 Overseas Institutions were co-operating. He wished to express appreciation to the Presidents and Councils of those organizations for their support.

A number of distinguished visitors from overseas were present, and he wished them a very hearty welcome to England and to the meetings.

The first stage of the scheme was completed. Before them there was a total of 136 papers, contributed by authors representing many branches of science and industry and from many parts of the world. Those papers were to serve as the basis of a general discussion which he trusted would, by contributing valuable further experience and views, prove a worthy complement to the papers. The resulting Proceedings, when published, would form a most valuable review of the present state of knowledge of that important and interesting field.

After explaining the method of procedure, the Chairman drew attention to the Exhibition which had been arranged at the Science Museum, South Kensington, to illustrate the technical state of the art of lubrication in its different fields and to interest the participants in the discussion and, to a lesser extent, the general public in modern developments in technique. The Exhibition would close on 31st October.

He then declared the meeting open to receive the papers of Group I, dealing with the subject of Journal and Thrust Bearings, and called upon Professor Swift, as Group Reporter, to deliver his summary of the papers of Group I.

Professor H. W. SWIFT (University of Sheffield) presented his report on the papers of Group I (p. 343).

Discussion

Professor G. B. KARELITZ (Columbia University, New York) remarked that a transitional stage had been reached in the development of the theory of lubrication in that it was necessary to look to the physicists again for further development. Bearings were made successfully before the theory of lubrication was investigated some fifty years ago. The application of theory to the actual development of bearings really began only about twenty years ago. Since then, as could be seen from the papers submitted for discussion, the theory had reached a very definite stage, and various practical questions could now be answered. It was possible to calculate the thickness of the film, and to estimate the friction losses rather closely. It was possible to find approximately what would be the side leakage of oil. In that respect, it was rather unfortunate that in one paper the so-called Karelitz formula for end leakage was used. When that formula was developed, there was nothing else available, but it was now obsolete, because the work of Needs and Kingsbury gave a much better answer to the question.

Before it became possible to give those cardinal answers regarding the thickness of the film, friction losses, and the amount of oil required, design had gone ahead of theory. The situation was somewhat similar at present. The thickness of the film could be found, but what was the allowable minimum? It was possible in automobile bearings to go as far as 0·0001 inch thickness of film under running conditions, with a given finish of the bearings and with given materials. No standards were available, however, for measuring finish, and there again, questions arose such as how to measure roughness, what was roughness, what were the characteristics of the surface created by running in a bearing? The temperature distribution throughout the bearing was not known. What happened near the film? What was the temperature of the shaft or of the brasses, and how should it be measured? What was the temperature drop and, in general, the heat transfer between the film and the metal? That was not known. It was for the physicist to give the information, and later on the engineer would apply it.

Information was lacking on a number of questions of design which were not covered by the papers presented. For instance, how to prevent leakage (where that was of importance) was not known, and no one knew what happened when there was a seal running on a shaft, as with certain aeroplane propellers. That was partly related to the question of what happened under a piston ring which rubbed on a wall.

The hydrodynamic theory was now sufficiently developed to allow of its utilization under most applications of steady load, but one was still at a loss how to employ it in the case of a variable load, as, for example, in internal combustion engines. It held there, of course, because otherwise the bearings would not run.

It was still necessary to reach some agreement on symbols and to simplify formulæ, tables, and diagrams in such a way that every designer could use them without having recourse to complicated computations. For instance, it was very important to reach, even in the expression of viscosity, some universal method of measurement, preferably, of course, by measuring the actual absolute viscosity of the oil.

Mr. W. A. Stanier (London, Midland and Scottish Railway Company) referred to a few problems which related to rolling stock. His paper indicated the direction in which English railway engineers were working, and it would be seen that the important factor for the locomotive bearing, as for any other bearing, was that it should be truly cylindrical and accurately finished. Many railways to-day were not only grinding journals but lapping them afterwards, in addition to providing the correct running clearance between journal and bearing and eliminating the variety of grooves which formerly existed in the bearing itself.

Those problems had been dealt with in a satisfactory manner on most railways.

An important problem was to prevent oil leakage and the ingress of water and dirt. If that could be solved satisfactorily, the number of "hot boxes" on the railways would be even less.

The viscosity of the oils used varied considerably. His paper indicated that for most purposes railways used a hydrocarbon oil blended with a certain percentage of rape oil, the percentage varying from 5 to 25, according to the experience gained. Provided that the oil was of good quality and the bearing was in good condition, he did not think it mattered very much whether a plain hydrocarbon oil or a blend of hydrocarbon oil with a vegetable oil was used.

Most bearings were made with whitemetal. The former practice was to depend upon the bearing to "lap up" the journal, a practice which he hoped and believed was being given up in favour of improving the mechanical finish of the journals. Provided both the journal and bearing were as near mechanical perfection as possible, and were given correct running clearance, no difficulty should be experienced with either locomotive or carriage and wagon bearings, there being, of course, a limitation on the unit load.

Mr. D. CLAYTON (Engineering Department, National Physical Laboratory) said that he could not altogether agree with Professor Swift's optimistic opinion of the agreement between theory and experiment in journal bearings, except in the general way one called descriptive. His experience on the experimental side had made him realize how great were the difficulties. In his paper, Prandtl claimed agreement with theory, but it was under conditions of thick film, where everything was favourable and differences were not critical. The difficulty arose when higher eccentricities were used. But, even with thick films for turbine bearings, Guy and Smith recorded marked differences between theory and experiment with regard to losses. Brillié claimed agreement of *his* theory with several sets of experiments, but found it necessary to use the results of experiment to choose amongst his three hypotheses, so that the check of his theory by experiment was not direct and independent. There might be nothing wrong with that procedure, but it was desirable to be sure that there was no suspicion of adaptability of the theory to suit each set of experiments because then there was no certain help for the designer.

He agreed with Professor Swift's remark that much of the experimental work of the past must be regarded as exploratory, but he did not feel that that comment should be restricted merely to the past. The difficulties arose because clearances were really so very small, and what

were really second-order small quantities became promoted to first order in importance. Small distortions, departures from true cylindrical form and so on, might make considerable differences to the results. The effects of those factors had become apparent with complete bushes, and it seemed inevitable that they would be worse with less rigid partial bushes.

Reading the papers confirmed his feeling that careful experimental work was urgently necessary. One outstanding problem, noticeable in a number of papers, was the determination of the running temperatures of bearings and the variation with operating factors, including heat loss factors. Without that information, the application of either theory or present experimental results was badly hampered.

If departures from geometrical truth were found in experimental work, they were likely to occur much more in practice in the making and setting up of bearings, and would be important in the choice of a factor of safety when satisfactory design data *became* available. He had worked up a result obtained by Jakeman and himself for a 360 deg. bearing and found an optimum eccentricity by Professor Swift's method which was about 0·7, compared with the 0·4 which Professor Swift gave for his 180 deg. bearing. Other results were similar, and he was, therefore, rather surprised that Professor Swift should be reluctant to make his optimum greater. Also, in the light of the papers presented, one might question the implication of Professor Swift's statement that the smaller eccentricity reduced the freedom of vibration, as apparently vibration in turbine bearings was less pronounced with greater eccentricity. The cause might be that the rate of change of eccentricity with load was less at greater eccentricity, so that the movement due to a disturbing force was less then than at lower eccentricity.

He had tried to understand the significance of Professor Swift's optimum eccentricity, but it was obtained by a computation from what appealed to him as a much more direct approach to the design problem, namely by way of the relations between film thickness and friction and the operating variables, of the form shown in Fig. 1 (p. 360).

McKee showed the use of the friction diagram alone, working from the minimum ; that was sufficient, with an appropriate factor of safety. A knowledge of the corresponding film thickness was useful, however, to obtain a figure which could be compared with the estimated accuracy of finish and setting up ; in fact, in helping to choose the factor of safety.

Regarding oil supply, Professor Karelitz, and Messrs. Juhlin and Poole showed how little oil was required for many applications. Some recent tests which he had made showed how badly it was possible to treat even a heavily loaded bearing after shutting off the oil supply without leading to failure. That was not surprising, since the leakage

of oil from the clearance space after drops had ceased to form could occur only by spreading along the journal or over the bush. In that respect, however, there might be some advantage in the complete over the partial bush. Those considerations of minimum oil supply helped to explain why Mr. Stanier's addition of 2 oz. per hundred miles might be sufficient for railway axle bearings. The contrast between that and the gallons per minute of the turbine bearing was very great, but apparently in the turbine bearing the cooling function of the oil had to be considered. It was interesting, however, to see from the paper by Messrs. Guy and Smith that a reduction of supply to a certain value resulted in a considerable reduction of losses, presumably churning losses. He was, however, surprised at the phenomena accompanying the reduction beyond that limit, apart from the temperature of the

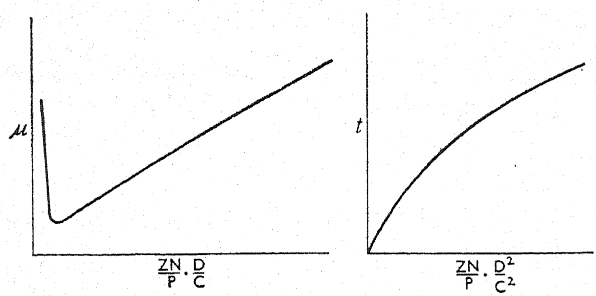

Fig. 1. Relation Between Film Thickness and Friction and the Operating Variables

bearing metal. Those peculiarities, and other information which those authors gave, justified their plea for extended, co-ordinated research on turbine bearings.

In that connexion, however, there was already the interesting paper by Mr. Samuelson. While, no doubt, the results were satisfactory from the point of view of eliminating fire risk, they were not less so from a wider point of view. Though a reduction of heat generation with reduction of viscosity was to be expected, not everyone would have dared to use a viscosity one-tenth of that of the usual oil. The bearing ran apparently satisfactorily at a higher eccentricity than usual, which was in the direction required for avoiding vibration. Further, the suction from the oil film itself was sufficient to supply lubricant to the bearing, provided the eccentricity was high enough. The water content would of course, give the water-base oil better cooling properties than plain oil. The use of thin lubricants did not appear to have received sufficient attention. It was usual to think of oil for lubrication, but other sub-

stances might serve the same purpose in particular cases. One of the indirect consequences of the use of bonded resins and rubber for bearings might be to emphasize that possibility.

A remarkable feature of the Discussion was the number of papers dealing with the failure of bearings. They were already familiar with the correction for eccentricity to the Petroff formula for viscous friction of a bearing. Now Professor Kyropoulos suggested (see vol. 2) that long-chain molecules in a liquid might produce "flow-orientation", leading to a decrease in viscosity of some 10 per cent. Mr. Brillié (see vol. 2) suggested that the formation of a boundary layer might lead to equivalent slip at the boundaries with decrease of friction, but also decrease of safety. Mr. Brillié and Professor Heidebroek considered the adverse influence of surface roughness locally on the pressure film formation. Mr. Brillié considered also that oil whirls might be set up in the tiny valleys of the surface roughnesses, giving an increase of friction. Messrs. Bradford and Vandegrift, and Mr. Needs, considered the effect of pressure on viscosity. It was going to be very difficult to sort out all those factors experimentally. Note might be made of the very small film thickness at which the pressure effect on viscosity became operative, say at about 0·00004 inch. Kyropoulos stated that hydrodynamic friction occurred down to at least 0·00008 inch. Heidebroek gave a figure of 0·000004 inch. Those figures were very small compared with the accuracy of even precise bearings.

Regarding running-in, it might be noted that, so far as the bearings were concerned, Mr. Stanier's procedure could be regarded as perhaps not the best way round. It would seem that the engine might be run quickly, as full film lubrication was obtained, and then be gradually slowed down to bring the surfaces gently into contact.

Mr. E. Watson Smyth (Taylor Brothers and Company, Ltd.) said that the introduction of fabric bearings to rolling mills had opened up an entirely new field to rolling mill engineers. If the practice of fitting fabric bearings to rolling mills was appreciated, the design of the mill should be arranged for the fabric bearings to be fitted in their best form. The method which consisted in simply fitting fabric bearings in place of the ordinary brass bearings was not satisfactory. Two experiments were mentioned in his paper, and it might be of interest to mention the results obtained. The first was to find whether the laminæ of the fabric should be laid vertically or horizontally. The bearing in the horizontal position rolled 7,640 tons before the strips had to be changed, while in the vertical position the first adjustment had to be made after 8,615 tons, showing a result roughly 1,000 tons better; but it so happened that, during the next thousand tons of rolling, the

strips failed altogether, and in the second experiment on the same lines, the strips failed after 5,000 tons had been rolled. It was fairly safe to say, therefore, that under heavy loads such as were met with in rolling mills, it was probably more satisfactory to have the laminæ in the horizontal position. He had been unable to carry out the second experiment on the use of fabric strips impregnated with graphite, because the manufacturers were uncertain about the exact performance of the material.

One of the chief troubles with fabric bearings in rolling mills was corrosion during shut-down. A fine, sandy material was formed on the shaft and if it was not dealt with, it would rapidly wear the bearings down. It might then be thought that one was not obtaining the proper life owing to the rolling, but that was not the case. The necks of the shafts had been coated with nickel, but that was a failure. If it were possible to obtain chromium-coated shafts in England, it would be a great advantage, but he knew of no plant in England which could coat shafts of the size required.

A matter which was causing great concern was the electrolytic action which took the form of very slight pinholes on the surface of the bearings. Everything possible had been done to overcome it, but so far, without very good results. That action took place on two mills, but the third mill was entirely free from it, and no reason was known why all three should not be affected. Any information on the point would be welcome.

Mr. H. Brillié (former Chief Engineer, Compagnie Générale Transatlantique) referring to the paper by Herr Falz, agreed that ordinary grooves of arbitrary design might be not only useless but harmful; but, if the design of the grooves was such as to permit of the continuity of the phenomenon of viscosity in agreement with Reynolds's formula, such grooves could be very useful as was indicated by theory and confirmed by experiment. Theoretically, the variation of pressure per unit of length depended only upon the thickness of the film at each point. At points where the thickness was greater than h', for which the parabola was a straight line, the variation of pressure was positive; on the other hand, the variation of pressure was negative for thicknesses less than h'. If the heights of a groove were greater than h', the increase would be less than for the film upstream, but the increase was always positive. With properly designed grooves, the pressure increased, but less quickly. The properly designed groove permitted of a new partial film, and the succession of partial films might be unlimited. On French ships where there were collars with properly designed grooves, the succession of films could exceed one metre in length. On the S.S. *Ile*

de France with properly designed grooves, very thick films of 0·3 mm. had been found instead of boundary lubrication such as was obtained without properly designed grooves. Without properly designed grooves as was indicated by Falz, the pressure film decreased quickly, and could have a length of only a few centimetres, as indicated by practice and from the experiments by Clayton, by Jakeman and Fogg, and by Thomson. Without grooves, one might have films of only a few centimetres instead of more than one metre with properly designed grooves. Such grooves ensured the formation of the film, security of running and efficiency.

Mr. F. SAMUELSON (The British Thomson-Houston Company) said that experiments with water-lubricated bearings were still continuing, and the results were very encouraging. He understood Mr. Watson Smyth to say that, with fabric bearings, the journals rusted. In his own case, there was no sign of rusting. When they opened up after a shut-down for two or three weeks, the journals were free from rust, and the bearing housing was remarkably clean; there was no sludge in it. They had found it necessary in starting up, if it were necessary to start slowly, to supply the lubricant under pressure to the journal, as, otherwise, there was trouble. He would recommend anyone interested to try the experiment; it looked very promising, and should be helpful in many cases.

Dr. R. O. BOSWALL (Manchester College of Technology) remarked that an unfortunate feature of the data given in papers relating to the film lubrication of the journal bearing was the lack of consistency with regard to symbols and units for the quantities appearing in the formulæ and curves. Direct comparison of results was, consequently, a difficult matter and the adoption of some standard system seemed to be necessary.

The operating conditions for a film lubricated bearing depended upon a criterion which involved load, speed and viscosity. The most frequently used form for this criterion was ZN/P, where Z represented viscosity in poises or centipoises, N denoted speed in revolutions per minute and P was the load in pounds per square inch of the projected area. Without expressing any opinion as to the suitability of the symbol Z—although it had been used for some years in America and in Great Britain—there did seem to be a definite advantage in adopting, for viscosity, the poise unit.

With reference to the data given in the speaker's paper, it had been assumed that the pressure film completely covered the full extent of the 120 deg. angle subtended by the brass. That meant that the film

conditions had passed through the transition stage—dealt with by Professor Swift *—when the film was incomplete owing to the production of negative pressures in the region of the outlet edge of the brass. It did not necessarily follow, therefore, that the bearing would not work effectively with values of $(ZN/P)(R/r)^2$ lower than those given in the tables, but simply that lower values for that criterion would lead to a reduction in the circumferential length of the film and reduced minimum film thickness.

Experimental data showing the relation between the pressure conditions and extent of the film for increasing values of ZN/P would be very useful.

The quantity of lubricant had an important influence upon the effective operation of the bearing since, if the quantity available was less than that corresponding to the value for $(ZN/P\ (R/r)^2$ and dimensions were as indicated in the paper (pp. 19, 20, 21), the bearing would come within the category of the starved bearings referred to by Professor Karelitz.

Mr. Guy and Dr. Smith mentioned (p. 119) a bearing 15 inches in diameter and 22 inches long which gave a loss of 109 kW. when supplied with 39 gal. per min. Calculation gave a minimum allowable quantity of 50 gal. per min. with a loss of about 65 kW. in the pressure film in the lower half of the bearing. The loss in the top half was not so easy to estimate, but it seemed probable that the total loss for that bearing should exceed 100 kW. It was noteworthy that the formula given by Karelitz (p. 151) showed that the side leakage for that bearing might be as great as 15 gal. per min. That was in agreement with a total quantity of 50 gal. per min.

The minimum film thickness for the bearing in question when film lubrication was perfect would be about 7/1,000 inch. The decreased loss that was observed with reduced quantity of oil could be attributed to the fact that the film did not attain that thickness owing to its being imperfect.

An important point concerning bearing operation that was frequently overlooked was that, due to certain peculiarities occurring in connexion with the position of the resultant film pressure, a bedded brass when centrally loaded would not work effectively under film lubrication conditions. It was, of course, common practice to use centrally loaded bedded brasses for railway work, and they operated quite satisfactorily. Conditions, however, were those associated with the greasy type of lubrication obtained when the supply of lubricant was

* "Stability of Lubricating Films in Journal Bearings", Proc. Inst. C.E., vol. 233.

scanty and it seemed probable that the successful operation of that type of bearing was largely due to the cooling effect of air circulation.

Film lubrication was a difficult subject and lack of exact agreement between theory and practice was inevitable owing to bearing surface irregularities resulting from inferior mechanical finish and deformation under load. Temperature effects might also have an important influence.

It was very desirable that everyone interested in the design or construction of bearings should make themselves acquainted with the fundamental principles controlling film lubrication since it was only by such means that the difficulties could be properly appreciated.

Mr. F. NIXON (Bristol Aeroplane Company) remarked that as an aero-engine designer he wished that research workers had investigated the behaviour of floating bushes. He agreed with Professor Taylor that the big-end bush of a radial aero-engine appeared to work almost in defiance of theory. Its design had been based largely upon empirical data. In the Bristol "Pegasus" engine, the big-end bush was floating, the mean bearing pressure was of the order of 3,700 lb. per sq. in., and in an Air Ministry type test, it had to stand a mean pressure rising to 4,700 lb. per sq. in. The point of application of the load moved over only a very small arc, as the major component of the bearing load was due to the centrifugal force on the relatively heavy master rod. The fluctuation of load, moreover, was very small. For the mean bearing pressure of 3,700 lb. per sq. in., the maximum was 4,700 and the minimum 2,700 lb. per sq. in. There were indications that fluctuation of load had a definite effect on the behaviour of a bearing. His firm's experience suggested that the smaller the degree of fluctuation the more arduous was the bearing condition. The extremely high pressures which could be successfully withstood by a small-end bearing were well known. There was a reversal in the direction of the motion of the bush relative to the pin, and that had been adduced as the reason for the ability of the "Pegasus" gudgeon pin bearing, for instance, to withstand maximum bearing pressures of 12,600 lb. per sq. in. without any trouble.

The Bristol big-end bush was made with numerous oil holes (48 holes in a bush $2\frac{3}{4}$ inches in diameter and 3 inches long), as they had found that they could not do without them. There was scope for considerable research into the problems of floating bushes. Increase of end leakage produced an improvement, probably because the oil was needed to remove the heat developed. It was more understandable when it was realized that, in the "Pegasus" engine, the whole of its 1,000 h.p. was taken by one big-end bush. The importance of rigidity and the bad effects of distortion, stressed by many of the authors, were amply confirmed by experience.

Dr. F. Bowden (University of Cambridge) said that most methods of measuring viscosity measured the viscosity of the oil in bulk; the resistance to flow of a relatively thick layer of oil was measured; but in the bearing itself, the layer might be very thin indeed. He referred to fluid lubrication, where the surfaces were separated by a very thin layer of lubricant, and it was of interest to inquire what was the viscosity of the oil when the film was very thin.

Many years ago, Professor Watson observed that apparently thin films of liquid, about 10^{-3} cm. thick, appeared to have a high rigidity. Hardy found some evidence for that as well, but Bulkeley filtered oil very carefully and measured its flow through fine capillaries of the order of 10^{-3} cm. and found no sign of rigidity. Some years ago, the viscosity of very thin films was measured at Cambridge * by making a viscometer of parallel plates which were separated only by half a wavelength of light, so that the thickness of the oil was of the order of 10^{-5} cm. to 10^{-4} cm. It was found that for most liquids—water, alcohol, paraffin, etc.—the viscosity of that very thin film was identical with that of the liquid in bulk. There was no sign of rigidity. The apparent rigidity reported by previous workers must have been due to dust or experimental error. That was true for most liquids. If, however, a substance which could form a liquid crystal was taken (fairly long molecules which had suitable hooks on the end or middle would hook together and form a chain), a different result was obtained. Thus, with water containing 1 per cent of ammonium oleate, the thin film which was 6×10^{-4} cm. thick had a very high rigidity indeed. A pressure of some centimetres of water could be applied to it, but it would not flow. Then, when the pressure was increased further, it began to flow quite quickly. If, therefore, substances capable of forming liquid crystals were dissolved in lubricating liquids or were initially present in the oil, the viscosity of the thin film might be very great. That high viscosity would not be detected by any of the standard viscometers since it only occurred when the film was very thin. Nevertheless it might play an important part in the starting and running of lubricated bearings.

Mr. H. L. Guy, F.R.S. (Metropolitan-Vickers Electrical Company) remarked that the General Discussion would achieve a useful and lasting piece of work if the co-operating institutions were asked to draw up a simple list of agreed symbols which could be used internationally in the literature. (Applause.) The task of studying the papers presented in Group I was at least doubled by the fact that, for instance, P sometimes meant load in pounds per square inch of projected area, when

* Boston and Bowden, Proc. Roy. Soc. A., 1935, vol. 151, p. 220.

that was not represented by p, or when it did not mean pounds per inch axial length of the bearing, while at other times it represented the total load on the bearing which, in certain papers, was represented by W.

He had been impressed by a remark (p. 139) by Messrs. Jakeman and Fogg, who, in reporting their results on seizing temperatures, said that in the earlier experiments, the results were erratic because the oil was not as completely filtered as in the later experiments. That comment went to the very heart of the difference between the results obtained in practice and those obtained in the laboratory, because there were three important factors in lubrication which controlled very considerably the performance of bearings. Those factors were dirt, air, and imperfections. He had been impressed again, by Mr. Clayton's remarks that a link might be found between some of the anomalies which engineers found in their work and the results of laboratory experiments, which had been very helpful but which had not yet gone as far as could be desired.

Another point of the same character occurred in the paper by Prandtl. Prandtl's paper was mainly concerned with the most satisfactory result that, once he sank his whole apparatus below the surface of the oil, the experimental results agreed with theoretical calculations. That, however, was the second stage in his work; the first stage being taken without the apparatus completely drowned, and then the results were very variable and erratic, and apparently gave friction coefficients, for instance, which were much higher than he found later. That, again, was a characteristic difference between so many engineering applications and laboratory work, because the engineer often could not drown his apparatus in oil, but yet he had to make it work. It might be suggested, therefore, that there were still a few ravines which had to be bridged, and it would be necessary to take the factors referred to into account in doing so.

Personally, he agreed that in order to control directly the design of apparatus dependent on the evaluation of friction, it was necessary first to adopt the most complete mathematical analysis available. It was then necessary to check that analysis by laboratory tests, but another step remained, for the engineer must then observe the ratio of the result obtained in practice to the predicted result of the analysis plus the laboratory correction, and the ratio of those figures would not be unity.

A remark which he would like to make about high-speed thrust bearings applied equally to journal bearings. He was confident that, if a designer would measure the actual losses in fifty installations under normal operation conditions, he would find that they varied on different jobs from once to twice the amounts deduced from published assess-

ments and he would find many more cases above 1·5 than he would find below it. The reason might once again be found in the inevitable dust, dirt, and air in the oil, and the imperfections of surface and dimensions in practical installations under operating conditions, when the temperatures varied in different parts.

He was sure that, if he asked engineers who were accustomed to seeing thrust bearing pads with unhardened collars after they had been in operation for some time, to sketch what they looked like they would draw them with an almost parallel scored band of narrow fine lines on the pad. Those fine lines were caused by the grit, the dust which was in the oil, but which caused no such scoring in the journal bearings of the same installation presumably because the minimum oil film thickness under the thrust pads was so much less than in the journals that dust became caught between the two and scored the thrust bearing, whereas it slipped through in the journal bearing. That condition, which was inevitable, might be one of the factors in the 50–100 per cent increase in the loss met in practice, compared with the limited laboratory test. The losses concerned were a small percentage of the total, but they were not insignificant in our national economy. Methods to avoid those losses, however, would be permissible only provided they were accompanied by increased security.

Professor H. A. Everett (Pennsylvania State College) said that he had paid a visit to St. Paul's Cathedral and had seen the wonderful mosaics there, and, while he had been listening to the present discussion, he could not help but draw a parallel between those mosaics and the programme which had been prepared for the General Discussion. There was a maze of gems, all interesting and attractive, assembled into a composite whole which made, to his mind, a magnificent presentation. He wished to express his appreciation of the courtesy extended to him and his thanks for permission to be present.

Mr. J. Foster Petree (Westminster, S.W.1) remarked that, towards the end of Professor Swift's report, it was stated that the Isothermos bearing circulated the oil by means of a disk. He believed that was not so; the Isothermos bearing had two revolving arms which picked up the oil and deposited it on to the top of the brass.

He would like to show some slides to amplify points not covered in his paper. Fig. 2 showed a Peyinghaus inner bearing for a locomotive axle. In his paper he referred (p. 238) to tests, at the National Physical Laboratory, of a multiple brass bearing formed with two, three, and four of the brasses shown. The bearing was made in one piece and cut away as required. The two-brass bearing need not be discussed. The

three-brass bearing would be used for a vertical downward load in the position shown in Fig. 3 or reversed, if the load was from below; in

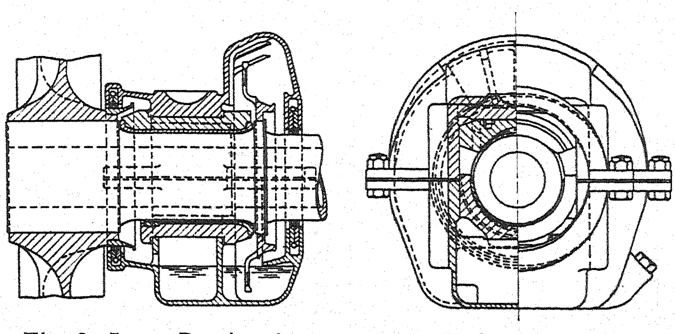

Fig. 2. Inner Bearing for Locomotive Axle (Peyinghaus)

every case there was a balance between the pressures from the different films. The four-brass bearing described in the paper was made with four equal segments. Segments of unequal size would be used, where,

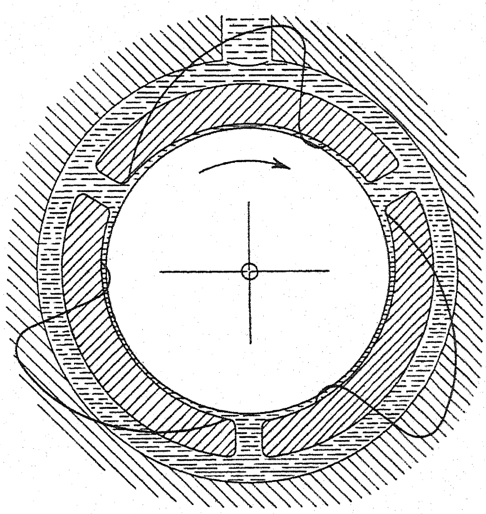

Fig. 3. Pressure Diagram for Three-Brass Bearing

for example, the maximum load was vertical and it was necessary to provide side steadiment. The films were, to all intents and purposes, balanced, the effect being that the maximum play was always less than

the design clearance of the bearing. There was a very definite centring effect, and in practice it was found that, although the clearance might be increased to twice or even three times the usual proportion, the shift did not vary *pro rata*; so that, for a given speed and viscosity, it could be arranged to have any oil film thickness desired, within a reasonable margin. The bearing was completely drowned at all times, and determined its own oil quantity. The tests at the National Physical Laboratory were purposely run with a much smaller oil quantity, but normally a considerable excess of oil would be provided.

The fact that (as mentioned in his paper) an important State railway system accepted as normal a variation in journal diameter of as much as 5 per cent due to wear, indicated clearly that lubrication by pad was not sufficient. Further evidence in support of that was provided, in that manufacturers of bearing metals were prepared to guarantee a minimum wear in a given length of time. Tests had been made on boxes of identical pattern with pad-lubricated and film-lubricated brasses to compare the running temperatures. In a series of 4-hour tests, with loads of 7 to 10 tons per bearing, and at speeds of 70 to 100 kilometres per hour, it was found that fluid film lubrication reduced the temperature rise above that of the atmosphere by more than 50 per cent.

He supported the suggestion that more attention should be given to the design of oil seals. He had tried various forms and, although many were very satisfactory with cold oil, they were by no means so with hot oil. In a particular case, where it was thought at first that it would be necessary to heat the oil artificially for the experiments, it was found sufficient to hold two oil seals on the shaft outside the bearing; all the heat required was provided by conduction along the shaft.

Mr. G. J. C. Vineall (Ragosine Oil Company), referring to the use of soluble oils in the lubrication of plastic bearings, remarked that it was known from the use of soluble oils in ordinary cutting that there were occasions where plastic caps swelled and distorted. Any soluble oil would not do, for it was necessary to have carefully chosen types of materials.

Herr F. Pesarese (Achslager-Syndikat, Berlin), commenting on Mr. Petree's paper, said that the greatest stresses imposed on an axlebox bearing were those due to axial shocks and pressures. These might amount to 70 per cent of the vertical load and, owing to the small end surface of the bearing brass, the intensity of pressure might reach 400 kg. per sq. cm. (about 5,690 lb. per sq. in.). The specific bearing load was only about 50 kg. per sq. cm. (711 lb. per sq. in.) on the projected area of the journal, and about 110 kg. per sq. cm. (1,564 lb. per

sq. in.) on the inner surface of the brass. It was desirable, therefore, to provide for axial shocks as fully as possible, and that could be done only by utilizing the whole of the collar surface of the journal. The construction shown by Mr. Petree (Fig. 3, p. 237) was noteworthy on that account, as the introduction of the under brass avoided a deficiency of collar surface in contact with the bearing. The under brass therefore conferred three advantages: (1) increased lubrication of the journal by means of the bath; (2) the bearing was safeguarded against displacement; (3) axial loads and shocks were safely carried by the joint action of the upper and under brasses, reducing the specific pressure on the surface of the journal collar by more than 50 per cent.

Professor D. DRESDEN (Utrecht, Netherlands) asked Mr. Samuelson whether the main reason for undertaking the very interesting experiments mentioned in the first part of Mr. Samuelson's paper was because disastrous fires had been reported.

Mr. SAMUELSON replied that that was so.

Professor DRESDEN remarked that the measure seemed a radical one, and inquired whether Mr. Samuelson was still optimistic, and believed that a practical use in central power stations of the cheap type of lubricant referred to would be possible.

Mr. SAMUELSON said that that was so.

Professor DRESDEN said that he was glad to hear it. He considered that that was one of the most interesting contributions, because it did away with many things which had previously been taken for granted.

Professor Dr.-Ing. H. FÖTTINGER (Institut für Strömungsforschung, Berlin) observed that it was usual to take the viscosity as a measure for the possibility of the application of an oil, but the general experience was that that physical figure was not sufficient, and so the conception of oiliness had been invented. So far as he knew, there was no concrete conception of oiliness, and therefore in the Charlottenburg Laboratory an effort had been made to apply the theory due to Reynolds and to extend it. His colleague, Dr. Vogelpohl, had found that some of the difficulties were solved by taking as a measure of oiliness the term $\beta/\gamma c$, β being a measure of the slope of the viscosity curve plotted against temperature. The slope of that curve had an important effect. Those oils which had a flat curve, e.g. the vegetable oils, gave lesser coefficients of friction for a certain load and for the same viscosity than

the mineral oils. The coefficient β was determined by the viscosity $\eta=\eta_1\times c^{-\beta(t-t_1)}$, η being the viscosity at a certain temperature and η_1 the viscosity at a fixed temperature. Where γ was the specific weight and c the specific heat, $\beta/\gamma c$ was apparently a measure of oiliness. The physico-chemical conception of oiliness took into account some properties like adhesion, which were not clearly defined, whereas that would be a purely mechanical conception. By analysis of tests made on the Thoma oil-testing machine, Dr. Vogelpohl had found that the friction observed for sugar solution, rape seed oil (crude and refined), and mineral oil depended directly on the $\beta/\gamma c$ values for different loads, as was shown by Fig. 4 of the paper by Dr. Vogelpohl (see vol. 2). It seemed to him that, on those lines, which were the lines of Osborne Reynolds, it might be possible to make an advance with regard to the very difficult conception of oiliness.

The CHAIRMAN said the organizers of the great gathering which he was addressing—and he referred in particular to Dr. Gough and Mr. Guy—could congratulate themselves on the most interesting discussion which had resulted from the first session. As Chairman, he proposed to say a word or two on the most interesting questions which had been discussed.

He would begin with Mr. Clayton, whom he understood to suggest that, if Mr. Stanier were to run his engines at express speed first of all when they were built, and then gradually run them more slowly, the bearings would be better bedded down. Personally, he knew that if an attempt were made to run a locomotive at full speed the day it was turned out of the works, it would not run very far before there would be blazing boxes all round. The only thing to do was to run them slowly at first and gradually bed them down. In theory, what Mr. Clayton suggested might be correct, but in practice it was not.

Mr. Samuelson had referred (p. 363) to the remarks of Mr. Watson Smyth regarding the bearings rusting, but Mr. Samuelson had probably not appreciated that Mr. Watson Smyth was talking about the very large bearings in rolling mills, which were surrounded by steam and water. If the rolling mill stood for a few hours, a little rust often formed on the bearings, and it was to that that Mr. Watson Smyth referred.

Dr. Boswall referred to bearings which, he said, would not work well if centrally loaded. Bearings used under railway carriages, however, were all centrally loaded. The only way to get them to run satisfactorily was by having very narrow bearings, and there were pressures up to 1,000 lb. per sq. in. Those bearings, if the carriage was stationary, and the wheels were revolved by means of rollers, would run hot immediately; it was the air cooling, of course, which kept them from

doing so in the ordinary way. He mentioned that, because those bearings were loaded, and must of necessity be loaded, exactly centrally. With over 100,000 carriage bearings on the railway with which he was connected, the return of "hot boxes" was often nil for a month, and in other months, there would be only one or two. That was proof that the centrally loaded bearing was not altogether unsatisfactory.

Mr. Nixon (p. 365) said that the floating bushes on Bristol engines had a very large number of holes through them. Professor Swift told him that the papers in Group I did not mention floating bushes. He thought that was an omission to be regretted, because floating bushes were very largely used in America in particular, and he had used them in this country, and remarkable results were obtained with them.

Mr. Guy (p. 366) had described the minute grooves in the pads of thrust bearings caused by grit, and said they were probably due to those pieces of grit being carried through. In railway practice, the converse was the case, and one had those minute grooves due to grit becoming embedded in the bearing, and it was quite common to have them on the bearing as well as on the pads.

Mr. Petree (p. 368) referred to the Isothermos disk, and said he did not think it was a disk. It was not a disk now, but when the Isothermos box was introduced twenty to twenty-five years ago, it was a disk. A number of them were still running on the London and North Eastern Railway after twenty or more years' service. The design had now been improved, but it served its purpose very well.

The Chairman then proposed a vote of thanks to the authors of papers, to the Group Reporter, and to all those who had taken part in the discussion, including in the vote the members of the Committee which had worked so hard to organize the General Discussion.

The vote of thanks was carried by acclamation, and the meeting then adjourned.

Communications

Mr. H. N. Bassett (Cairo, Egypt) wrote, with reference to the paper by Professor Karelitz, that, while loss of oil by end leakage was undesirable, it was even less desirable that effectiveness of lubrication should depend upon the bearing heating up to a point at which the viscosity was sufficiently reduced for the oil to rise satisfactorily through the waste (p. 151). Ideally, a film of oil should be formed with the first complete revolution of the axle, but if high viscosity oils were used, that could not occur, or would be unlikely to do so if the axle had been stationary for any length of time. An epidemic of hot boxes on wagons

in Egypt some years ago (lubrication being by pad oilers, fed by worsted lickers from oil contained in the bottom of the box), was due to the use of an oil which was far too viscous at starting temperatures to flow through the worsted. Since an oil of lower viscosity was used, hot boxes had been fewer, and pads were rarely damaged, whereas, at one time, damage to pads was a frequent occurrence. It would be useful if Professor Karelitz would suggest suitable viscosities for oils for waste-packed boxes under (*a*) sub-tropical and (*b*) temperate conditions.

The viscosity range quoted by Mr. Dowson for the oil used by Maker *g* in Table 1 (p. 77) appeared to be far greater than was desirable, as the corresponding range at starting temperatures would be much wider, so that an unnecessary amount of power might be wasted in starting.

The use of so much lead in the whitemetal of Maker *c* appeared to be questionable. It cheapened the alloy, but in the unlikely event of the lubricant supply failing, the leaded whitemetal would break up under load more quickly than would an alloy free from lead. Both the antimony and the copper contents in the whitemetal used by Maker *f* appeared to be excessive. Experimental work* indicated that antimony in excess of 8 per cent in whitemetal of that type made little difference to the hardness. An excess of copper tended to cause brittleness in the alloy.

The addition of copper to the lead-base alloy mentioned by Professor Kraft on p. 165 (lead 73, antimony 16, tin 10, copper 1) was not necessary, as it made very little difference to the hardness, and also had but little effect upon the yield point of the alloy. Ackermann † had shown that the addition of copper to that type of alloy did not confer increased wear resistance. The introduction of copper led to the formation of extremely hard copper-tin needles, which were also brittle and easily displaced from the matrix. In thin linings, e.g. in thrust bearings, copper was said to lead to brittleness. On the whole, the alloy was better for the purpose for which it was quoted without copper. The antimony content was also rather high for the particular application envisaged, as though the hardness was increased the alloy tended to become brittle. A better composition would be 72 lead, 13 antimony, and 15 tin.

The suggestion by Mr. Macnaughtan (pp. 187, 383) that cycles of heating and cooling of a local area of the surface of a bearing would lead to cracking of the lining, was interesting. In use, however, whilst there was local heating, even to the melting point of the eutectic (shown by the

* Ellis and Karelitz. Trans. A.S.M.E., 1930, vol. 52, MSP–52–7.
† Zeitschrift für Metallkunde, 1929, vol. 21, p. 339.

speckled appearance of bearings at times, when dismantled, the speckling being due to the wiping of the molten eutectic), it was doubtful whether there was sufficient *difference* of temperature involved to lead to cracking. The temperature of the oil in the bearing was not so low as might be imagined, and hence the cooling effect referred to was minimized. The suggested reason for cracking might explain isolated cases, but did not seem to be generally applicable. There was ample evidence to show that faulty adhesion of the lining to the shell was an important contributory cause of cracking. The writer had secured virtual freedom from cracking in railway wagon brasses by paying special attention to adhesion.

The figures given by Mr. Stanier on p. 307 showed that English railway companies used considerable amounts of rape oil in locomotive journal oil, in one case as much as 25 per cent. Experiments made by the National Physical Laboratory years ago on the effect of the addition of rape oil to mineral oils used in the Lanchester worm gear showed that no appreciable benefit was derived when the proportion of rape oil in the compounded oil was raised above 2·5 per cent. Therefore, the use of as much as 25 per cent of rape oil appeared merely to add to the cost of lubrication without making any appreciable difference to its efficiency; 5 per cent would be ample. The improvement depended upon the proportion of free fatty acids present. Possibly the company using 25 per cent of rape oil insisted upon an oil almost completely free from free fatty acids when blended, in which case the larger proportion of rape oil would be required. By making the specification for the rape oil less stringent, less oil would be needed. Fatty oils always tended to increase in free fatty acid content during use, and, for that reason, limitation of the rape oil content to 5 per cent would appear desirable.

The statement (p. 335) by Messrs. Welter and Brasch that the use of aluminium and zinc alloys as bearing metals would be affected by the introduction of high-pressure lubrication, suggested that the hardness of such alloys was the sole reason for their lack of employment. That was not quite true, for, with alloys containing 92 per cent aluminium and 8 per cent copper used in experimental railway bearings, it was the practical difficulty of bedding the bearings to the journals which led to the abandonment of the experiments. Heating was frequent and wear excessive. One of the main objections to the use of zinc alloys as bearing metals was that they could not be cast as linings to shells, but must be used as the complete bearing. Zinc alloys containing free zinc (i.e. zinc not combined with any other constituent of the alloy) tended to seize readily on steel, causing scoring of the bearings. Possibly, that defect would be less marked with high-pressure lubrication.

Whilst the writer agreed with Mr. Petree that oil bath lubrication

gave the best results (p. 234), it was not a practical proposition in railway practice generally for plain bearings. The very satisfactory results given by pad oilers indicated the efficiency of the method of lubrication adopted. Experimental work was done on coaches fitted with axleboxes incorporating a dipper for raising oil from a well in the bottom of the box to the top of the brass. It was found that the boxes ran hotter than others fitted with ordinary pads, and the wear on the brasses was excessive. That was due mainly to the abrasive action of the tin-antimony cuboids torn out of the matrix by dust which passed the dust shield, both cuboids and dust being maintained in circulation in the oil. The metallic matter found in the oil after ten days amounted to as much as 0·6 per cent, mainly tin-antimony crystals. In the normal type of pad-oiled bearing, such cuboids (extremely hard and abrasive) remained at the bottom of the box when they were loosened at all, and the wear of the bearing was much less than with the splasher type of oiler because only clean oil, freed from abrasive matter by passage through the worsted lickers and the pads, came into contact with the journal and the brass.

Mr. Dicksee stated (p. 70) that, with ordinary tin-base materials, temperatures in excess of 100 deg. C. would not be tolerated for more than occasional short periods, and the inference was drawn that the reason was that linings cracked readily at above this temperature. The actual reduction in ultimate tensile strength of a high-tin tin-base alloy on heating to 100 deg. C. was roughly 40–45 per cent, but of more importance than the reduction in tensile strength was the reduction in hardness. The metal tended to spread in the bearing, clearances thereby being much increased. That was more important than tendency to cracking, which arose more from faulty adhesion than from reduction in tensile strength, though the latter was a factor.

Dipl.-Ing. P. BEUERLEIN (Rhenania-Ossag Mineralölwerke A.-G., Hamburg) wrote to give some further results of the researches described in his paper.

Greases. The same tests as those illustrated in Fig. 1 (p. 9) were carried out in baths at 50 deg. C. Swelling began much sooner, and after 200 hours was greater than that recorded in Fig. 1 after 20 days. The greases were found to follow the same order as before. Greases 1–7 were spread on bushes which were then kept for 200 hours at 80–85 deg. C. The bushes showed a permanent contraction of 0·2 to 0·22 mm. in external and internal diameter. These bushes were then placed in water baths at 20 deg. C. After 7 days, the swelling was greater than that of the heated bushes after 20 days. It therefore appeared that an increase in temperature from 20 to 50 deg. C. greatly increased the swelling.

Bushes which had been heated at high temperatures for a lengthy period showed greater tendency to swell.

Oils. The shrunken bushes were placed in oil baths at 20 deg. C. Although the previous work showed that prolonged heating increased the tendency to swelling, no alteration in dimensions was observable after 6 weeks. When the temperature of the oil baths was raised to 50 deg. C., the bushes began to swell after about 100 hours. Steady values were attained after 350 hours, the figures at 450 hours showing no change. Pure mineral oils and compounded mineral oils showed no difference in their effects, though sulphurized oils had a surprisingly smaller influence on the swelling. The extent of swelling appeared to depend on the viscosity of the oil. Oils with a viscosity over 12 deg. Engler at 50 deg. C. produced a swelling of slightly less than 0·1 mm. on the external diameter after 350 hours, whereas, with a lower viscosity, the swelling was greater. Oils 7 and 12 showed an increase in the external diameter of roughly 0·32 mm. It thus appeared that oils with a viscosity of less than 3 deg. Engler at 50 deg. C. should not be used to lubricate fabric bearings.

Comparison of the results obtained with oils and the other lubricating and cooling media, showed that the oils had the least influence on swelling; at room temperature, no swelling could be observed with the oils. Fabric bearings lubricated with oils could therefore be given smaller clearances, provided that the bearings were properly worked.

Fabric bearings made from fine cotton fabric ("Hartgewebe F") and pressed to shape were next tested. The external and internal diameters were 40 and 17·96 mm. respectively and the length was 25 mm. Tests were made at 50 deg. C. in water (*p*H 7·9), oil 9, and a 1/20 emulsion prepared from the emulsifiable, corrosion-preventive oil. In water and in the emulsion, swelling began after 2 days; in the oil bath, after 3 days. Final values were reached after 3 weeks. The internal diameter decreased by 0·3 mm. in water, 0·2 mm. in the emulsion, and 0·08 mm. in the oil. Water again produced most swelling, while emulsions from suitable oils caused somewhat less swelling than water, though the difference was trivial. The best behaviour was shown by the pure oils.

Mr. H. BRILLIÉ, replying to the remarks by Mr. D. Clayton, wrote that no theory could provide a complete solution of the problems of viscosity in bearings, as the limiting conditions for the integration of Reynolds's equation were not known. That was why there was disagreement between experimental results and many theoretical conclusions, as the latter were only arrived at by means of hypotheses which, as experiment showed, corresponded only to particular cases. The three conditions, A, B, and C, which were considered by the writer,

were not hypotheses but really three special cases from which practical cases could be arrived at by interpolation. The mode of interpolation could only be decided by experiment. The advantage of that method was that the elementary theory of viscosity was utilized to the fullest extent, while it was left for experiment to ascertain, gradually and with increasing strictness as development ensued, what was lacking to complete the technique of lubrication.

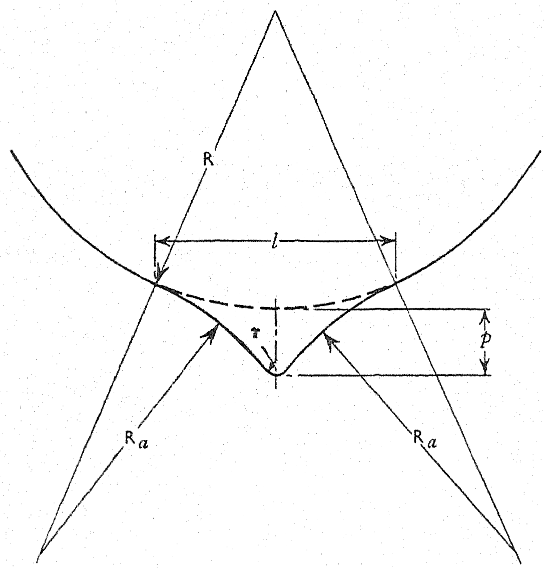

Fig. 4. Diagram of the Brillié Oil Groove

R radius of brass. R_a radius of nozzle.
l width of oil pocket. r fillet.
p depth of oil pocket.
$p \times 2R_a = l^2/4$ (approximately). In general $l = R/5 = D10$, l being usually >4 mm.

In reply to the observations by Herr Falz (p. 380), Mr. Brillié wrote that his remarks on transverse grooves applied to all grooves with sections of such a design as to ensure the continuity of hydrodynamic phenomena. Experiment had shown that satisfactory results were given by grooves having a section formed by two arcs of a circle tangential to the cylindrical section of the brass and having the greatest possible radius compatible with the depth desired, the two arcs being joined by a fillet r of small radius (Fig. 4).

The addition of the grooves in no way affected the length required for the bearing surface and only ensured greater stability. The grooves enabled greater length to be obtained, if desired, while restricting the passive resistances as far as possible. The writer considered that the prime qualities of an oil film were its stability and its thickness, a guarantee of safety, as in practice the film was always liable to be too thin, so that conditions of boundary lubrication were approached whereas such conditions should be avoided if for any fortuitous reason the viscosity of the oil might become too low.

Mr. E. B. Davies (Bakelite, Ltd.) wrote that, in the paper by Mr. P. Beuerlein, mention was made of only one material having been tested. There were various other types of synthetic resin material for bearings and those different materials, if subjected to similar tests, would probably show very different results. The swelling of synthetic resin fabric material in the presence of water was well known, and if water was to be present in the lubricant, that slight growth in size could be allowed for in the clearance. When bushes were required for accurate work involving a fine clearance, certain grades of material could be given a pre-treatment which would nullify or at least largely diminish the swelling referred to by Mr. Beuerlein.

Referring to graphite-impregnated material, attention must be paid to Mr. Eyssen's remarks with regard to electrolytic action between graphite and steels in the presence of water, more especially if the steels were nickel-chromium alloys. It was doubtful whether graphite had any beneficial effects in reducing friction or assisting lubrication when water was used as the lubricant.

It was suggested in Mr. Rochester's paper that the best method was to mould the fabric bearing so that the load was applied to the flat of the material. The load then came on the length of the fibres and not on their ends. Some experiments showed, however, that better resistance to wear was obtained when the load was applied on the end of the fibres. It appeared that, with the fibres lying flat, embedded in the synthetic resin, they would not give up or hold the lubricant to the same extent as when end on.

In some papers, lists of physical properties of certain synthetic resin bonded materials were given, without anything to indicate the wearability of the material, either with or without lubrication. As a manufacturer of synthetic resin bonded material for the production of bearings, the writer's firm was primarily concerned with the wearing properties of those materials and in carrying out tests on two different grades of material, both of which had physical properties as indicated in the papers under discussion, it was found that one wore away very much faster than the other under similar test conditions.

The necessity for some indication of permissible bearing pressure was becoming more and more apparent. The generalization that fabric replaced bronze satisfactorily was far too broad. Failures through excessive wear had been experienced on high speed mills, but splendid results had been obtained on low speed mills, such as strip mills cold rolling stainless steel. One such bearing showed little wear after a year's continuous service, even though it was estimated that the bearing pressure in the neck exceeded 5,000 lb. per sq. in. In that particular bearing, the lubricant was a water soluble oil, and while water alone could be used successfully as a lubricant, experience indicated that the slightest presence of oil miscible with the cooling water, if it were permissible, was more than just helpful.

Successful installations which had been operating for some time indicated that, while the cost of synthetic resin bearings was appreciably more than bronze bearings, the production life was greatly increased, so that, considering first cost against production, synthetic resin bearings were the more economical. Further economies were power savings per ton of production, with consequent increase in production, while grease lubrication charges were reduced, stoppages for changing bearings were more infrequent, repairs to roll journals were eliminated, consistency of gauge of rolled metal was held more accurately, and there was no necessity for frequent stoppages to allow journals to cool.

Mr. G. R. Eyssen (Hydro-Plastics, Ltd.), in a written reply to Mr. E. B. Davies, pointed out that the graphite materials were generally for dry running operations or possibly for use with solvents having a poor lubricating action. In mentioning the electrolytic action between graphite and steel in the presence of water, it was not meant that the water should be used as an assistant lubricant, but that the moisture of the atmosphere and condensed atmospheric humidity were sufficient to act as an electrolyte, especially when the sulphur content in industrial atmospheres and the salt content in coastal air were taken into account.

Obering. E. Falz (Hannover-Kleefeld) wrote that the remarks in his paper referred to oil grooves of the customary type. The remarks of Mr. Brillié were thus not clear in the absence of a knowledge of the design of the grooves which were claimed to give the advantages in question. Further, it was not stated whether such long layers of lubricant were necessary to take up usual loads. If that was not so, an excessive length of oil film implied an unnecessary loss of power. The length of the oil film was not a criterion of the goodness of a bearing, which was shown by the lowest possible coefficient of friction.

Mr. H. L. Guy and Dr. D. M. Smith wrote that while they appreciated the interesting point raised by Dr. Boswall (p. 364) in connexion with the tests reported in Fig. 1 of their paper (p. 120), they doubted whether the bearing in question came within the category of "starved" bearings. Dr. Boswall had overlooked the fact that in a completely enclosed bearing a considerable amount of recirculation occurred in the oil. From his own figures based on Karelitz's formula 15 gal. per min. would leak sideways. Therefore, for instance, on that basis if 40 gal. was supplied to the bearing, 25 gal. per min. was being recirculated. That difference emphasized the authors' point of the undesirability of experimenting with partial bearings instead of the complete bearing which was common in practice. The authors would also point out that if it was established that an imperfect oil film was obtained with the smaller quantities of oil used in the experiments, the immediate deduction would be that complete bearings worked satisfactorily with a less loss with partial oil films than with complete oil films. The authors, however, doubted whether Dr. Boswall's argument was valid, because in the series of tests of which one was reported, oil quantities were supplied considerably in excess of the 50 gal. mentioned by Dr. Boswall. On the bearing reported in the paper, oil quantities were used up to 66·5 gal. per min. and in all tests the losses were found to increase with the oil quantity. In another series of tests in which the bearing diameter was 12 inches and the length 16 inches, again running at 3,000 r.p.m. and with the same total clearance of 0·030 inch on the diameter, the variation in oil quantity covered the range 6·8 to 69·5 gal. per min. Hydrodynamically, that range of quantities corresponded to a variation from 11·7 to 119 gal. per min. on the 15-inch bearing. In the last-named tests it was also found that the loss increased continuously with the oil quantity, but that the rate of increase became very small for the 12-inch bearing above 50 gal. per min., a quantity which corresponded to 86 gal. per min. on the 15-inch bearing.

Mr. F. Hudson (Research and Development Department, Mond Nickel Company, Ltd.), in a written contribution, referred to the paper by Mr. Macnaughtan, principally dealing with the failure of tin-base bearing metals due to fatigue, and pointed out that, whilst the addition of cadmium effected some improvement, the increased fatigue resistance was apparently not enough to overcome the trouble positively in many practical applications. It was doubtful whether the modified alloy could, as yet, equal the new cadmium-nickel alloys which had a tensile strength of just over 2·5 tons per sq. in. at 150 deg. C. compared with 2·25 tons given in the paper for the modified tin-base bearing metal and just under 2·0 tons for the straight alloy. Possibly the addition

of a small amount of nickel in conjunction with cadmium to the tin-base metal might effect still further improvements in the resistance to fatigue failure. A straight alloy containing about 0·6 per cent nickel * had been reported successful in aircraft work.

In many cases, fatigue failure of bearing metals could only be avoided by using the copper-lead alloys, since their physical characteristics, as pointed out by Messrs. Neave and Sallitt (p. 210), were less affected by temperature. Aircraft engines, almost exclusively, and many automotive engines to-day, carried bearings of that alloy. It should be pointed out, however, that those alloys were very sensitive to cooling conditions during manufacture, and must be chill-cast or cooled fairly rapidly to obtain the best results. An alloy containing 85 per cent copper and 15 per cent lead had only a tensile strength of about 4 tons per sq. in. with 5 per cent elongation when sand-cast, against 6 tons and 8 per cent, respectively, when chill-cast. A strength of at least 5 tons was maintained in the latter condition at 150 deg. C.

Copper and lead would not alloy in the true sense of the word and, consequently, were very liable to segregate. It had been stated that additions of other metals were theoretically prejudicial to thermal conductivity, but it was considered that the benefits obtained, such as greater homogeneity, outweighed most other disadvantages. Furthermore, it was debatable, in view of the thin bearing section of the metal employed, whether thermal conductivity played any part at all. Some comments had been made on the effect of nickel in that direction. There were probably three reasons for the action of nickel in preventing lead segregation. First, nickel caused the metal to set more quickly, a distinct advantage as chilling or quick setting was essential. Second, an addition of nickel to copper actually increased the solubility of lead in molten copper. Whereas pure molten copper could dissolve only about 38 per cent lead, the addition of, for example, 2·5 per cent of nickel enabled it to take up 60 per cent or more. That ensured the absence of segregation and a finer dispersion of lead globules. Third, if copper-lead alloys high in lead were overheated, there was a tendency for the liquid emulsion to be destroyed, so that they could not be chilled or cooled fast enough to prevent serious lead segregation. The addition of nickel stabilized the emulsion at high temperatures. A marked difference in lead dispersion with and without nickel in those alloys was observable under the microscope, particularly with 1·0 to 1·5 per cent nickel. Such a composition was being successfully used.

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* Sutton, H. Jl. Inst. of Metals, 1934, vol. 55, p. 97, "Discussion on Bearing Metals".

In Germany, where the bearing problem was affected by restrictions on copper and tin, advances had been made in the use of copper-lead by modifying the design so as to develop the full load carrying capacity of the alloy, with a consequent saving of 25 to 30 per cent in weight of metal required. Such bearings did not "run themselves in" and must be finished to size by fine boring, preferably with a diamond tool. For that, and other reasons, generous clearances were required, normal German practice in engine bearings being to use a clearance of 0·001 to 0·0015 times the journal diameter.

Systematic investigation of all bearing metals was required. In fact, indiscriminate additions of other metals were dangerous unless the resulting alteration in properties was fully known. For example, recent investigations indicated that additions up to 3 per cent of nickel to the bearing bronze containing 80 per cent copper, 10 per cent tin, and 10 per cent lead, resulted in a progressive improvement in tensile strength, limit of proportionality and yield point with some reduction in ductility. That alloy was of value where maximum strength was required, but, if maximum toughness was desired, by limiting the addition to 0·5 per cent an all-round improvement in properties was obtained, the ultimate strength being increased by 8 per cent, the yield point by 5 per cent, the limit of proportionality by 3 per cent, and the elongation by 22 per cent. That increase in ductility *in conjunction with improvement in the other properties* resulted in a definite advance and indicated further possibilities. Such a modification was of value for high duty bearings, particularly when resistance to "pounding" was desired.

In conclusion, it would have been useful to have heard something about the bearing properties of the aluminium- and zinc-base alloys. Early attempts to utilize the aluminium base alloys had not met with general approval and their latest development * would be followed with interest. The zinc-base alloys had proved of value in machine tool work, and in many cases had given excellent service. An alloy containing 83·5 per cent zinc, 10 per cent copper, and 6·5 per cent aluminium had been used for bearings in automatic machines for the mass production of small turned articles.

Mr. D. J. MACNAUGHTAN (International Tin Research and Development Council), with reference to Mr. Bassett's comment (p. 374) on the suggestion that cycles of heating and cooling of a local area of the liner might lead to cracking of the whitemetal, pointed out that it was important to distinguish between bearings carrying more or less steady loads and other bearings carrying loads that pulsated rapidly. It was

* Rolls-Royce, British Patent No. 470,248.

in the latter case only that cracking occurred in the peculiar manner in question, by tessellation, and it was under such pulsating loads only that cyclic changes of temperature were considered to afford a possible explanation of the cracking. In the more common case of more or less steady loading, it was agreed that cyclic variation of temperature was unlikely to occur in any degree that could lead to cracking.

The importance of adhesion between the whitemetal and the shell (referred to by both Mr. Bassett and Mr. Dicksee) was recognized in all cases—alike for steady and for pulsating loads.

A systematic study of various factors that influenced the adhesion between tin-base bearing metals and different liners was in progress under his direction, and it was hoped to publish results at an early date.

Messrs. D. P. C. NEAVE and W. B. SALLITT (Copper Development Association), replying to Mr. F. Hudson, wrote that they were fully in agreement with his remarks (p. 382) concerning the beneficial influence of nickel on the casting properties of copper-lead alloys; but they regarded as less acceptable his view that thermal conductivity played little or no part in copper-lead bearings on account of the thin bearing section of the metal employed. An aspect of the question which appeared to have been overlooked was the importance of securing the rapid diffusion of heat from the local "hot spots" caused, for example, by a temporary breakdown of the oil film. Owing to the low thermal conductivity of the steel shell, the thin section of the copper-lead lining clearly imposed a restriction on the flow of heat from a locally overheated area, and therefore accentuated rather than obviated the need for an alloy of high thermal conductivity. Though by no means easy, it was commercially possible to cast copper-lead bearings without additions of nickel or tin, and the performance of such bearings appeared to warrant the extra care required in their manufacture.

Professor F. K. G. ODQVIST (Royal College of Technology, Stockholm) wrote that as the wording of the second paragraph in his paper (p. 227) was rather misleading, he wished to make it clear that it was not his intention to diminish the well-deserved recognition due to Mr. A. G. M. Michell for his invention of the bearing with multiple tilting pads. Osborne Reynolds could claim to have originated the idea of a single pad bearing, the theory of which was developed in the "Philosophical Transactions" of the Royal Society for 1886. The splitting-up of the bearing surface into a series of pads was, as far as he himself was aware, due to Mr. A. G. M. Michell and Dr. A. Kingsbury, working independently of one another.

Mr. W. A. STANIER (London, Midland and Scottish Railway Company), writing with regard to Mr. Foster Petree's remark (p. 370) that an important railway system accepted as normal a variation in journal diameter of as much as 5 per cent due to wear, thus indicating fairly clearly that lubrication was not sufficient, replied that the figure quoted was very high and would not be recognized by English railways. For carriage journals the figure should be not more than 1 per cent and for wagon journals not more than 3 per cent.

Professor H. W. SWIFT (University of Sheffield) wrote regretting that his views regarding the proper hydrodynamic basis for journal bearing design had not proved acceptable to Mr. Clayton (p. 358), who proposed to base design on the value of (ZN/P)(D/C) giving the minimum coefficient of friction on a certain curve on the grounds that McKee "showed" that to be a sufficient basis for determining the optimum eccentricity.

McKee had in fact, more cautiously, "suggested the possibility of obtaining a basis of design" by substituting in a heat dissipation equation a limiting value of (ZN/P) chosen with a reasonable margin of safety above the minimum point of curves of μ plotted against (ZN/P). In other words, McKee realized that that minimum point was a danger point and not an optimum.

In his "much more direct approach to the design problem" Mr. Clayton proposed to use relations between μ and (ZN/P)(D/C) and between the film thickness t and $(ZN/P)(D^2/C^2)$. Again more cautious, McKee had stated that the product (ZN/P) "determines the value of both the coefficient of friction and the relative film thickness for a given bearing". That was, of course, true "for a given bearing", i.e. for a bearing whose dimensions D, B and clearance C were already fixed. But to take a "given bearing" and find a suitable duty for it was hardly in the best traditions of design and to apply to one bearing the relationships between μ and (ZN/P)(D/C) or between t and $(ZN/P)(D^2/C^2)$ determined from another having a different D/C ratio was contrary to the principle of dynamical similarity.

In the problem presented to the designer the load, speed and lubricant were usually fixed and the choice of journal diameter was often restricted or dictated by other conditions. He had a little more control over the axial width, but the feature over which he had most control was the clearance, which must be regarded as the ultimate element in design.

Until the essentials of design were fixed, and particularly the clearance, it was necessary to employ more fundamental relationships than those contemplated in the "direct approach" method, which only

applied to a "given bearing". The proper generalized friction relationship, for example, was that between $\mu(D/C)$ and $(ZN/P)(D^2/C^2)$. Consequently if μ was plotted against ZN/P or $(ZN/P)(D/C)$ each clearance ratio would have its own curve and the minimum value for μ on a particular $\mu : (ZN/P)(D/C)$ curve would not in general give the least value of μ obtainable.

Generalized relationships useful for design purposes existed between any of the basic dimensionless quantities: $\mu(D/C), t/D, \epsilon, (ZN/P)(D^2/C^2)$. From those, equally general relationships could be derived between $\mu\sqrt{P/ZN}$ and ϵ and between $t/D\sqrt{P/ZN}$ and ϵ. For theoretical conditions those relationships were plotted * and showed that for given load, speed, and lubricant, optimum frictional and film thickness conditions were obtained if the clearance was so chosen as to give the relevant running eccentricity shown in Table 1, p. 310, i.e. to give the corresponding value of $(ZN/P)(D^2/C^2)$.

For practical conditions when side leakage had to be considered the numerical relationships were modified but the basic procedure was unaffected and was fully developed in the Proceedings of the Institution, vol. 129, 1935. In that treatment, for want of evidence to the contrary, it was assumed that the optimum value of the eccentricity ratio was unaffected by side leakage. Subsequent data provided by Mr. Needs and reinforced by the case mentioned by Mr. Clayton suggested that side leakage tended to increase the value of the optimum eccentricity. If that were fully substantiated it would involve certain numerical adjustments in the design formulæ, but it so happened that the operating conditions were not very sensitive to the chosen eccentricity ratio between values of 0·35 and 0·75 and the smaller the eccentricity ratio adopted the closer would be the fit of the bearing and, *pace* Mr. Clayton, the less the freedom for vibration and the greater the range of safety for efficient operation.

Dr. A. S. T. Thomson (Royal Technical College, Glasgow) wrote that the agreement between Mr. Brillié's theoretical analysis and the experimental investigations carried out at the Royal Technical College, Glasgow, was very interesting. In the experiments, the curtailment of the effective bearing arc with decrease in the value of ZU/P was quite considerable, which was in accordance with the latest theoretical treatment. That feature demonstrated the futility of using bearings of large arcs of embrace, as such bearings only led to greater power loss without a commensurate gain in load capacity.

He was glad to see that work was being done on the influence of

* Swift, H. W. Proc. Inst. C.E., 1932, vol. 233, p. 284, Fig. 10.

pressure on viscosity. It was suggested that therein lay the explanation of oiliness. Possibly it did, to a certain extent, but the property of oiliness, as influencing static friction, was apparent in the Deeley machine at loads as low as 10 lb. per sq. in. With regard to the upturning of the friction curve at low values of ZU/P which, Professor Bradford submitted, was due to increase in viscosity with pressure, there was evidence that that upturn was associated with high point contact of the surfaces.

Turning to the use of fabric bearings, in tests run at the Royal Technical College, Glasgow, considerable swelling of the material was observed. It was interesting to have confirmation of this in Beuerlein's paper. Taking swelling into account, together with the low viscosity of the lubricating water, and the very heavy loads in rolling mill practice, it was not thought possible that viscous film conditions could exist in those bearings, but that the lubrication was of some boundary form.

Was there not some error in Mr. Rowell's paper where it was stated (p. 267) : "It is also well known that if the journal is accurately fitted to the curvature of the shaft, the film of lubricant is of even thickness, can be maintained at maximum thickness and lowest coefficient of friction . . ."? That would appear to be contrary to hydrodynamic theory and experience.

In conclusion, he hoped that one of the results of the Discussion might be the standardization of nomenclature and units. At present, much work was involved in interpreting the figures for various investigations due to the wide range of units and symbols employed.

GROUP II. ENGINE LUBRICATION

OIL CIRCULATION SYSTEMS IN THE LUBRICATION OF INDUSTRIAL MACHINERY

By Lt.-Colonel S. J. M. Auld, O.B.E., M.C., D.Sc.,* and
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For the purpose of this paper the term "oil circulation system" is intended to apply to the lubrication of the moving parts of prime movers or machines by continuous circulation of oil. It is not concerned with actuating systems or with those used purely for cooling, but does apply to those important cases where the oil is used to carry away frictional or conducted heat from the bearing which is being lubricated.

Circulation systems are incorporated in many classes of machinery. In many cases, as in large pumping arrangements, the train of gears forming the transmission is fully enclosed and separately lubricated by circulation.

Compounded oils are seldom used in circulation systems. The presence of the compound adversely influences both chemical and physico-chemical stability. The mineral oils which are mostly employed are chosen from such stocks and refined in such a manner as will (*a*) allow them to withstand as far as possible the severe conditions to which they will be subjected, and (*b*) at the same time guide such changes as are inevitable along the least harmful lines.

The two factors tending to change the composition of mineral hydrocarbon lubricating oils in use are oxidation and heat. These factors almost always act together, but the heat effect and the oxidation mass action vary greatly according to the conditions. Thus in a turbine the temperature is low and the heat influence small, whilst the oxygenation is extensive and the direct oxidation effects marked. In an internal combustion engine, on the other hand, the maximum temperatures encountered may be high and the direct oxidation effects reduced to a minimum by suitable engine design.

There is a close similarity between the course of decomposition of those petroleum fractions which constitute the lubricating oils under the influence of heat and of oxidation at elevated temperatures respectively. This similarity allows the course of alteration of the oil

* The Vacuum Oil Company, Ltd.

in use to be followed chemically with some success and has correspondingly important influence upon the design of both engines and machines. The relationship can be explored with advantage in a variety of typical cases. In each there are a number of common factors based (1) upon thermal and oxidation behaviour, and (2) upon the accumulation of impurities. The latter may strongly influence the course of the former, catalytically or otherwise, but in any case must be eliminated or minimized because of potential mechanical interference with the supply and application of the lubricant.

Most of the factors affecting the design of the lubrication system can consequently be classified as in Table 1.

TABLE 1. FACTORS AFFECTING DESIGN OF LUBRICATION SYSTEM

Factor	Mode of action	Influence on design
Oxidation . .	Aeration . .	Pipe and tank design; type of pump; pressure and rate of circulation
	Catalysis . .	As under impurities
Heat . . .	Temperature effect .	Temperature control; design of coolers; design of heaters; control of conducted heat
Impurities . .	Emulsification and sludging	Prevention of water intrusion; design and layout of settling tanks; installation and use of strainers, filters, and centrifuges; removal and treatment of oil

The effects on the oil which it is wished by choice of grade and design of machine to minimize are chiefly: (*a*) increase in viscosity; (*b*) formation of solid deposits (including carbonization); (*c*) certain types of acid formation; and (*d*) sludging and loss of ability to separate from water. Any or all of these phenomena may adversely affect the ability of the oil to carry out lubrication and heat transference. The actual happenings in the oil may be pictured with some degree of probability. This picture can be used to guide the choice of type of crude oil from which the lubricant should be prepared and the best method and extent of its refining.

The changes in the oil are more or less gradual and are of two kinds: formation of solid condensation products and rise in neutralization value. This is the initial course taken by the oil whatever the height of the temperature in the heat-oxidation factor. The condensation products may be purely acidic in type, but are much more likely to be polymers or condensed nuclei at the same time. They may be

soluble or insoluble in oil. It is the soluble products which are responsible for increase in viscosity. Both are liable to occur together and both may result in objectionable semi-solid deposits and difficult emulsions. In addition the insoluble products may form deposits in the pipe system and in oil ducts and other passages.

Tendency towards solid deposition has been largely eliminated by choice of suitable oils and design. In its place, however, over-refining produces an equally objectionable tendency to acid formation of a type which may result in corrosion and the formation of metallic salts and soaps. Copper salts thus formed are strongly catalytic and consequently "snowball" the oxidation effect. Iron salts and soaps hydrolyze, giving colloidal iron hydroxide. In the presence of soluble oxidation products which act as dispersing agents this solid is adsorbed at an oil-water interface and produces bad emulsions. These emulsions are sometimes so recalcitrant that the oil has to be discarded.

"Inhibitors" or anti-oxidants are intended to retard the oxidation processes, but care must be exercised in the choice and use of such materials, as their limitations are not yet wholly known and many have marked practical disabilities. In the meantime the best results have been obtained by controlled refining and the choice of oil stocks which will give the least harmful oxidation products.

The effect of heat and oxidation on oil at higher temperatures is chemically more obscure, but the results are more obvious. Gumming and the formation of carbonaceous deposits are only too well known. In internal combustion engines where the same oil is used for bearings and cylinders the effect will be most marked in the cylinder heads and behind the rings. It is immaterial whether lubrication is by splash or force feed so long as the sump is the source of the oil. It is indeed likely to be in the crankcase that the damage is done by initial oxidation of the oil in atomized form. Such oxidized oil is highly susceptible to decomposition in the cylinders. The course of events is then really a continuation of the oxidation condensation which may have resulted in the thickening and sludging of the oil in the crankcase. Under the higher temperature in the cylinders there is continuous loss of hydrogen from the hydrocarbon molecules by further polymerization or even by "cracking", so that in time insoluble carbonaceous materials are formed and, in the end, carbon.

It is apparent, therefore, that the chief problems connected with oil circulation systems are (1) the provision of an oil of maximum chemical and physical stability; and (2) the design of the system to ensure the gentlest conditions of usage for the oil and simplify its maintenance in a clean condition. Reference may be made to a number of typical cases.

Enclosed Steam Engines. Oil circulation systems fitted to these units frequently encounter difficulties not met in other types of machinery. One of the most objectionable is the intrusion of water from piston rod and valve spindle glands, as such water, on finding its way into the crank chamber, carries with it steam cylinder oil, mostly in the form of emulsion. A very small percentage of this emulsion adversely affects the separating qualities of the oil in the crank chamber. Such oil-water emulsions are not only harmful in themselves but hasten oxidation by reason of the largely increased oil surface exposed.

Between the bottoms of the cylinders and the top of the crank chamber there is in most engines a space or gland pocket. In some of the older types of engines it has been the practice to lead drainings from these spaces to so-called oil-and-water separators external to the crank chambers. Whilst such a design allows the separation of suspended oil from the water and its return to the crank chamber, it is also the means of introducing cylinder oil into the chamber. To obtain the best operation it is advisable to collect oil from the oil-and-water separator and ascertain by examination whether it is contaminated or suitable for re-use. Frequently much of the oil is that which is removed by the oil scraper gland at the top of the chamber. It is unfortunate that these oil scraper glands are occasionally badly designed and may actually assist the oil to escape.

Another feature for control is the height of the oil level in the crank chamber. This should be such as to prevent the crank webs and moving parts from dipping into the oil and setting up a churning motion which may assist oxidation. In the presence of water, oxidized matter and extraneous material this may result in the formation of bad sludges. It is for similar reasons that oil pressures in crank chamber lubrication systems should receive attention. In most engines 5 to 10 lb. per sq. in. is ample. Pressures above this only result in excessive leakage from the bearings and consequent atomization of the oil. Finely atomized used oil is liable to deterioration in the presence of moisture in a heated atmosphere.

Steam Turbines and Similar Systems. Aeration of an oil not only increases the mass of the oxygen with which it comes in contact but greatly increases the surface exposed. It is therefore one of the chief factors in the acceleration of oxidation and is similar in its effects to oil atomization or mist formation. In all circulating systems it is desirable, therefore, that subdivision of particles and air entrainment be minimized. Nowhere is this more important than in a steam turbine. It is next to impossible to eliminate aeration entirely, but it can be much reduced: (*a*) by the employment of suitable circulating

pumps; (*b*) by streamlining the pipe work; (*c*) by preventing splashing and cascading of the oil; and (*d*) by removing unavoidably entrained air quickly.

The valveless ram pumps often employed for oil circulation in high-speed steam engines are quite satisfactory because they are submerged in oil. Otherwise, ram pumps are the least suitable type for oil circulation because of their liability to encourage aeration. In general the most suitable type is the rotary pump, provided with efficient glands or sealing arrangements to prevent ingress of air. Such pumps give a continuous flow, with a comparative absence of surging. Circulating oil pumps are often fitted with de-aerating devices. These pumps also maintain a constant pressure regardless of changes in oil viscosity. Pump efficiency may be seriously impaired if care is not given to ancillary arrangements. A typical case open to criticism in this way was that of a centrifugal pump supplying oil to a gravity tank feeding the gears and bearings of a large papermaking machine. The gravity tank was fitted with a float control to close the discharge when the oil reached a predetermined level. On closing the discharge a spring-loaded relief valve lifted and released oil on a bypass to the suction side of the pump. The result of this arrangement was bad churning and serious overheating of the oil.

Pipe work throughout should as far as possible be in straight runs with gradual bends. Sudden changes in direction are to be avoided. At points such as returns to the main tanks the oil should not be allowed to fall from any height; the oil should either flow down a ramp of suitable slope or else the return pipe should carry the oil just below the working level in the tank. The former method is preferable and should be installed if space permits. The thinner the film and the more surface exposed the greater the opportunity for disengagement of entrained and occluded air.

Closely connected with this effect is the rate of circulation. Whilst this must be governed primarily by the exigencies of temperature control the designer must allow for sufficient oil to be in circulation to give it ample time to free itself from air, or for froth to disperse whilst it is in the main tanks.

The bulk and velocity of the oil are also intimately connected with the settling of moisture and solid impurities. Although secondary, these factors are of as much importance to the behaviour of the oil as oxygenation itself. Table 2 gives the quantities and composition of typical deposits recovered either from turbine system settling tanks or centrifuges with different oils in use.

The extent to which these deposits are composed of metals and metallic compounds is to be noted. The catalytic effect of such materials on

oxidation has already been emphasized. It is not possible, therefore, to exaggerate the importance of their removal during use and hence of the need for good settling and separation. Prevention, however, is better than cure. The presence of free metal and metallic oxides in the early stages of use may seriously affect the future history of a turbine oil. Far more attention should be paid to the initial cleaning of new systems than is often the case.

Once deterioration has started the finely divided mixtures of water, oxidized oil, metal and metallic compounds, and impurities which

TABLE 2. COMPOSITION OF TYPICAL DEPOSITS FROM DIFFERENT OILS

Make and size of turbine	30,000 kW.	20,000 kW.	15,000 kW.	7,000 kW.	25,000 kW.	20,000 kW.	10,000 kW.	7,500 kW.
Carbonaceous materials, per cent	16·5	9·3	25·9	25·6	33·2	7·3	35·92	32·2
Oil, per cent .	17·6	51·9	44·6	36·3	28·9	80·07	35·70	57·2
Water, per cent .	39·5	24·37	3·0	33·6	35·0	8·55	21·70	6·0
Ash:—								
Silica, per cent	1·8	1·4	2·8	0·5	0·8	0·37	0·36	0·3
Iron oxide, per cent	23·7	11·1	17·4	1·6	1·2	1·68	1·92	1·8
Copper oxide, per cent	—	1·2	6·3	1·8	0·5	1·56	0·69	0·2
Calcium oxide, per cent	—	—	—	0·2	0·1	—	Trace	Trace
Zinc oxide, per cent	—	0·3	—	0·1	0·1	0·44	2·21	2·3
Sulphate, per cent	0·9	0·4	—	0·3	0·2	0·1	—	—
Chloride, per cent	—	0·1	—	—	—	—	—	—
Period of use of oil, months	9	74	1	41	62	56	54	(?)

collect in all circulating systems offer media all too good for the auto-catalysis which results in the self-poisoning of the oil. Unless tanks are suitably designed, centrifugal separation alone is not sufficient to ensure the removal of such material. The tanks must be easily accessible for cleaning and with sloping bottoms to facilitate deposition and drainage.

It is not unusual for oil tanks to be contained in the cast-iron sole plates of the machines. Such tanks almost invariably have flat bottoms and frequently have ribs which extend to a height of 3 or 4 inches and form traps for water and sludge in positions which it is impossible to

drain. In some cases it is not even possible to gain access for hand cleaning after the oil has been removed. Fig. 1 shows a typical case embodying such difficulties. Fig. 2 in diagram gives a modern layout in which drawbacks of this nature have been obviated.

In all large oil circulating systems it is necessary to have the oil under efficient thermal control. The temperature of the oil coming to the bearings governs the heat exchange and consequently the required rate of flow and with the latter the volume of oil needed. On the

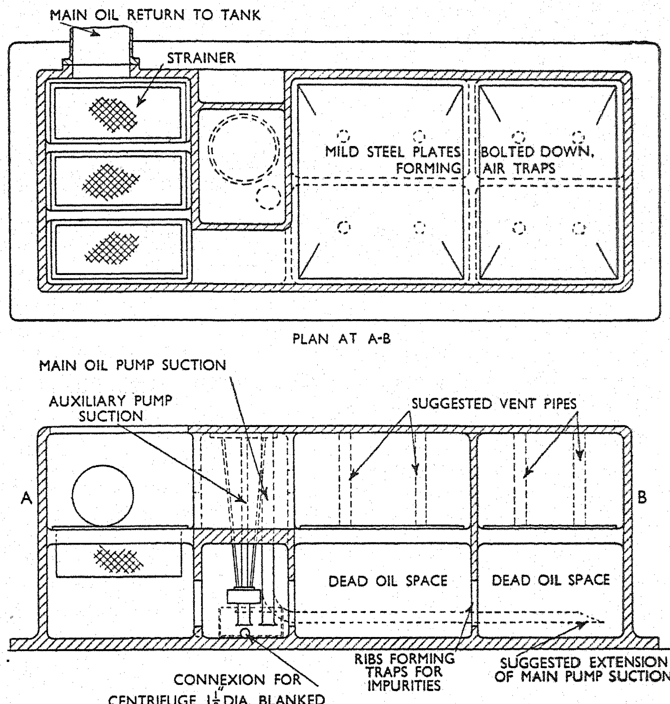

Fig. 1. Unsuitable Design of Oil Tank

other hand the oil should be sufficiently warm to permit of ready separation from water. Where centrifugal separation is effected the best results are obtained with the oil as cool as is compatible with efficient handling. This applies equally whether the treatment is by means of a bypass or whether the whole oil is treated on shut-down. The reason is that some of the solid products of oxidation redissolve in warm oil. In theory, therefore, the separation of water and the separation of solids should be effected at different temperatures. In practice, space and design may not permit the installation of the

necessary tanks, although certain of the most modern arrangements do incorporate a form of combined cooling and settling tanks. Close check must be kept on the temperature control of the coolers. Undue cooling of the oil is thus to be avoided, because oil containing water together with dissolved and suspended impurities may then deposit an emulsion on the tube surfaces. Not only does this impede heat transference but such emulsions are frequently the forerunners of heavier sludgy deposits in the tubes or elsewhere.

Most of the foregoing remarks apply generally to circulating systems for turbines. The chief difference from systems incorporated in rolling and paper mills, and so on, is the higher viscosity of the oil.

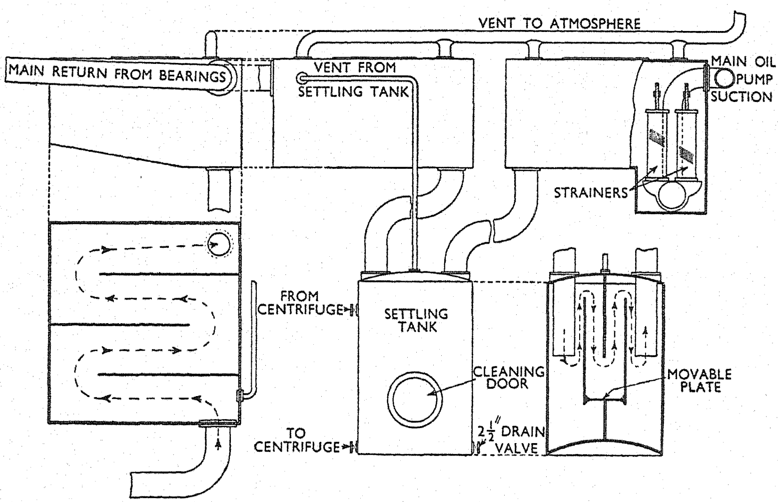

Fig. 2. Suitable Design of Oil Tank

The heavier the oil the greater the difficulty of separation from water and other impurities and of maintaining good demulsibility. Contamination with condensed moisture and process material is frequent, so that good settling is essential, and the use of internal steam or electrical heaters may be necessary to warm the oil and facilitate the separation of water. They are also useful at starting, when the heavy-bodied oil might circulate with difficulty.

Internal Combustion Engines. Conditions in the oil circulation systems of internal combustion engines are more severe than the foregoing. The oils themselves also are generally of higher viscosity and molecular weight, and more susceptible to heat-oxidation. In

addition they are more difficult to refine to the point where they will separate readily from moisture and other impurities. It becomes of ever greater importance, therefore, to pay attention to the points of design and operation already discussed, i.e. pumps and pumping; limitation of atomization and aeration; and removal of impurities and deposits.

Prevailing temperatures in circulating systems of internal combustion engines may be generally higher than those of turbines or machine units, but the real trouble occurs where a single oil is used for bearings and cylinders. Not only is the crank chamber oil liable in this case to contamination by products of combustion from the cylinders, but partially oxidized oil from the crank chamber when splashed or fed to the cylinders will decompose more quickly than fresh oil.

The efficiency of the lubrication of the cylinders of an internal combustion engine may be largely gauged by the condition of the piston rings. Oil trapped in the ring grooves must withstand resinification and carbonization, otherwise the rings may set solid, with disastrous results. It is apparent therefore that the circulating oil must have the highest possible thermal stability. Even "oiliness" may have to take second place to this requirement.

The systems themselves may test an oil severely. Ducts and passages are intricate and of small dimensions, as well as frequently changing in direction. Steady flow of oil is essential if deposition is to be avoided, particularly at sharp corners and edges. Indirectly, such conditions may also result in mist formation and aeration.

The circulating oil in internal combustion engines gradually becomes contaminated with carbon from fuel oil, with its own decomposition products and with adventitious matter. In large engines some form of constant filtration is very desirable. In small engines the question does not arise so often since the oil can be changed as required at no great cost. The practice of installing filters integral to the engine is becoming more general. In engines of moderate power containing, say, upwards of 150 gallons of oil, the filter is usually on a bypass on the oil circulation system. The best practice is undoubtedly to remove the whole charge and treat it at one time; this, however, calls for a double supply of oil and separate filtration equipment.

In large engines, where piston cooling is effected by oil circulation, conditions are especially severe. Oil is not a particularly good cooling medium, and if its flow is interrupted at high temperatures it will crack and form carbonaceous material, thereby still further restricting the flow. This vicious circle may be started by aerated oil reaching the pistons and reducing the rate of heat transference. Special care must be taken to prevent this from taking place.

THE PETROL ENGINE AND ITS FRICTION

By R. Barrington, M.Sc., and J. L. Lutwyche, B.A.*

The motor vehicle is one of the largest users of lubricants; probably at this moment in Great Britain alone there cannot be much less than 4,000,000 gallons of oil in circulation. Strangely enough, engine friction was regarded as a necessary evil for many years, and until comparatively recently very little progress had been made towards a complete understanding of the somewhat intricate problem. Gradually, however, more and more knowledge is being acquired, and there is every hope that one day conjecture will give place to certainty.

The authors' experiences of engine friction have been gained while assisting in the development of starting equipment for motor vehicles. It is common knowledge that engine resistance is a maximum when starting from cold, and it is this particular aspect which has received the closest attention. A previous paper † covers earlier work, and it is the authors' intention here to set out briefly some of the more salient features of investigations which have since been carried out on problems of engine friction. It is proposed to deal with the subject under the following headings:—

(1) The static or "breakaway" torque.
(2) The cranking torque.
(3) The running torque.
(4) The starting of an engine.

(1) *Static or Breakaway Torque.* In starting an internal combustion engine the static or breakaway torque must first be overcome. This torque varies with the size and tightness of the engine and with the position of the pistons in the cylinders. Opportunity occurred for measurement of the static torques during the re-assembly of a four-cylinder engine of 1,650 cu. cm. at −1 deg. C. Readings were taken every 30 deg. of engine revolution, after standing for at least 5 minutes in the required position. The crankshaft alone was first explored and then the pistons and connecting rods were assembled without rings; next the two types of ring—compression and scraper—were tried, first separately and then together; finally, the camshaft, oil pumps, distributor, etc., were added. The lubricating oil in this instance was a straight mineral oil of medium viscosity.

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* Joseph Lucas, Ltd.

† Jl. Inst. Automobile Eng., 1935 May, vol. 3, No. 8, p. 15, "Ease in Starting Petrol Engines from Cold".

Fig. 1 shows these static torques under the various conditions of assembly, plotted against crankshaft angle. It should be remembered that an engine almost invariably stops with the pistons approximately half-way up the stroke, i.e. where the static torque is highest. It is of interest to note the comparative effects of the two types of piston ring, compression and scraper. The latter, by the nature of its design, has a smaller effective area, and hence the pressure per unit area is considerably greater. The addition of two compression rings per piston increased the maximum static torque from 5·7 lb.-ft. to 7·8 lb.-ft., an increase of 37 per cent, and the further addition of the two scraper

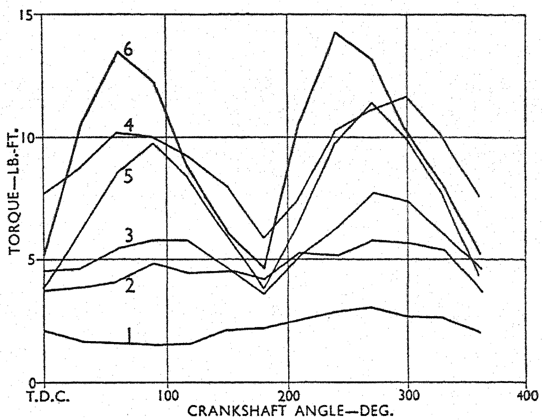

Fig. 1. Static Torques under Various Conditions of Assembly, plotted against Crankshaft Angle

1 Crankshaft only.
2 Plus pistons, no rings.
3 Plus compression rings only.
4 Plus scraper rings only.
5 Crankshaft and complete piston assembly.
6 Engine complete.

rings to each piston increased the maximum static torque to 11·7 lb.-ft., an increase of 50 per cent.

(2) *The Cranking Torque.* As a result of earlier investigations, it was stated that the relationship between cranking torque, speed, and oil viscosity could be closely expressed by

$$T=A+B\sqrt{NV}$$

where T is the crankshaft torque in pound-feet, A and B are constants depending upon the engine, N is the speed in revolutions per minute, and V the viscosity of the oil in centipoises.

Some effort has since been made to obtain further knowledge of the factors influencing A and B, and as a result it is now considered that the relationship should in fact have been expressed simply as $T=K\sqrt{NV}$. It is believed that as the speed decreases this formula holds good down to the limiting value originally expressed by A, and that at this point the friction is no longer fluid. The difference in the calculated curve of engine resistance, resulting from the use of the corrected formula, is very slight, except over the lowest range of speed. Tests have been made on an engine at −1 deg. C. (with the plugs removed and the camshaft disconnected) down to 8 r.p.m. (when varia-

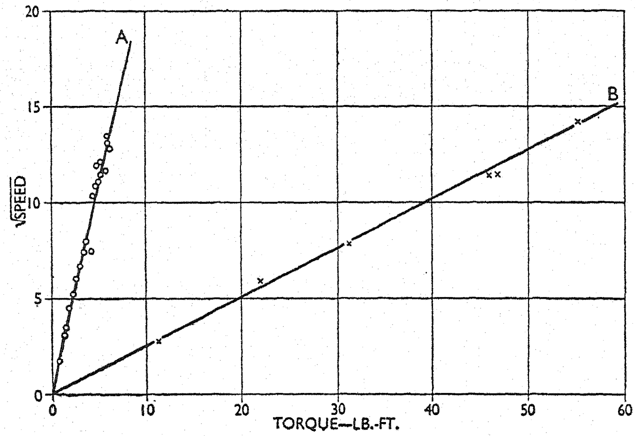

Fig. 2. $\sqrt{N}$–Torque Curves for Crankshaft and Engine

A Crankshaft of 1,500 cu. cm. four-cylinder engine ($T=0{\cdot}46\sqrt{N}$).
B 1,650 cu. cm. four-cylinder engine ($T=3{\cdot}9\sqrt{N}$); plugs removed and camshaft disconnected.

Viscosity constant at 2,500 centipoises.

tion in speed during a revolution complicated the issue) and on a crankshaft assembled alone in its bearings down to 3 r.p.m. (when there was some evidence of the oil film failing); and as will be seen from Fig. 2, there is every indication that, provided there is no change in conditions, the curves connecting torque and $\sqrt{N}$ for constant viscosity are actually straight lines passing through the origin.

Some of the factors influencing the value of K have been investigated. Records of a very large number of cold tests at −1 deg. C. on cars of various makes and sizes were examined and the K constant was plotted against engine capacity without much success. It was then plotted against swept area, as the major part of the friction is

located in the cylinders; the results, shown in Fig. 3, may be of some use when forecasting the approximate resistance curve of an engine whose bore and stroke are known. It will be seen that, very roughly,

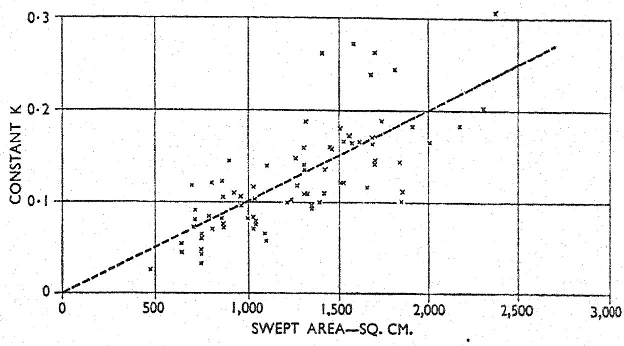

Fig. 3. The K Constant Plotted against Swept Area

the constant amounts to 0·1 per 1,000 sq. cm. of swept area. In each of these cases the oils concerned were the usual winter oils recommended by the manufacturers.

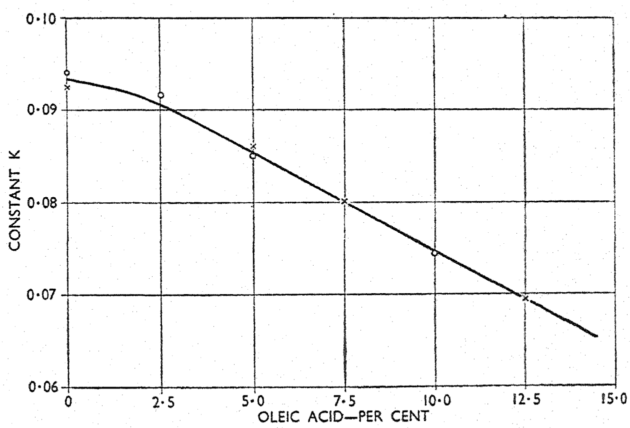

Fig. 4. Effect on the Value of K of Adding Oleic Acid

○ Medium engine oil. × Light engine oil.

The effect upon engine friction of the addition of various percentages of oleic acid has been examined. It was found that a very definite change in the K constant followed the addition of this "oiliness" agent, K decreasing almost linearly with increasing percentages of oleic

acid. Tests were carried out on a medium engine-oil, and a light engine-oil. The maximum addition of oleic acid was 12·5 per cent and the viscosity was measured at each addition. It will be seen from Fig. 4 that there is a drop of about 2·5 per cent in the value of K for every addition of 1 per cent of oleic acid. The static torque is only slightly affected and falls to approximately 85 per cent of the initial value at the first addition of 2·5 per cent oleic acid, and does not decrease with further additions. The "just moving" torque, i.e. the torque which just maintains motion at the lowest speed possible, is quite unaffected. It appears from this that K is also dependent upon the nature of the lubricant. In practice, however, it has been found that there is very little variation in K with the usual motor oils, even when compounded oils are used; presumably because the percentage of compounding matter is small.

The growth of K as an engine is assembled was examined at the same time as the static torque. The partly assembled engine was motored at various speeds and the torque plotted against $\sqrt{NV}$ to obtain the various values of K. Table 1 shows the variation of K under various conditions of assembly, together with the calculated torques necessary to motor the assembled parts at a normal cold-cranking speed of 75 r.p.m. with a medium winter oil having a viscosity of 2,500 centipoises at —1 deg. C.

The effects of the two types of piston ring upon the constant K are

TABLE 1. CRANKING TESTS ON SUB-ASSEMBLIES OF A 1,650 CU. CM. FOUR-CYLINDER ENGINE

Condition	Formula constant K	Calculated crankshaft torque at 75 r.p.m. and viscosity 2,500 centipoises, lb.-ft.
Crankshaft only	0·021	9·3
Pistons: no rings	0·079	34·2
Pistons: scraper rings only	0·085	37·0
Pistons: compression rings only	0·090	39·0
Pistons: both rings	0·100	43·5
Camshaft added	0·101	43·7
Big ends tightened	0·104	45·0
After running on load	0·094	40·7
2½ per cent oleic acid added	0·091	39·3
5 „ „ „	0·085	37·0
7½ „ „ „	0·080	34·8
10 „ „ „	0·075	32·4
12½ „ „ „	0·069	30·1

very similar, the compression rings causing a slightly greater increase, due to their greater effective area of contact with the cylinder walls. The camshaft appears to add very little; final tightening up of the big ends added 3 per cent, but the first run of the assembled engine reduced the torque by 10 per cent. The final torque was then reduced by 20 per cent by adding 10 per cent of oleic acid and running to circulate. These results are also shown diagrammatically in Fig. 5

(3) *The Running Torque.* The use of the formula for calculating engine resistance curves at high speeds and temperature has been

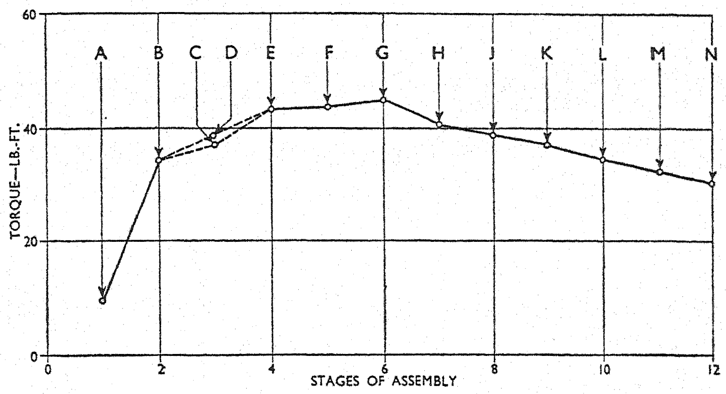

Fig. 5. Cranking Tests on Sub-assemblies of a 1,650 cu. cm. Four-cylinder Engine

Speed, 75 r.p.m.; viscosity, 2,500 centipoises.

A Crankshaft only.	H Run under load.
B Pistons, but no rings.	J 2½ per cent oleic acid in oil.
C Scraper rings only.	K 5 ,, ,, ,,
D Compression rings only.	L 7½ ,, ,, ,,
E All rings.	M 10 ,, ,, ,,
F Camshaft.	N 12½ ,, ,, ,,
G Big ends tightened.	

tried with promising results. A six-cylinder engine of 2,400 cu. cm. capacity was cranked at −1 deg. C. and the relationship obtained. It was then mounted on an electric dynamometer, run to get thoroughly warm under its own power, and then as quickly as possible the dynamometer was set to motor the engine which was switched off and the throttle closed. This was done at various speeds up to 3,000 r.p.m. Taking the oil temperature at the working faces to be approximately that of the water jacket, the points obtained fall very close indeed to the calculated line, as will be seen in Fig. 6. Had the exact

temperatures of the oil on the cylinder walls and at the bearings been known, it is at least probable that the high-speed test points would have been even closer.

(4) *The Starting of an Engine from Cold.* Considering a normal 2-litre engine whose friction torque can be expressed by $0{\cdot}12\sqrt{NV}$, running with a winter oil in the sump, having a viscosity at -1 deg. C. of 3,600 centipoises, it can be deduced that the power required to overcome its internal friction at 500 r.p.m. (a reasonable "tick-over" speed under cold conditions) is approximately 15 h.p. At 500 r.p.m. the indicated horse-power of an average 2-litre engine is unlikely to exceed

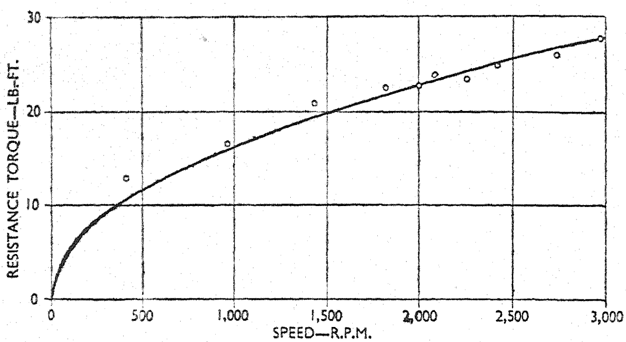

Fig. 6. Predicted Engine Resistance Curve and Actual Measurement

2,400 cu. cm. six-cylinder engine. Curve deduced from formula $T = 0{\cdot}128\sqrt{NV}$ based on cold tests. Marked points obtained on dynamometer. Oil temperature, 85 deg. C. Viscosity, 15·8 centipoises.

12, so that it would not at first sight appear to be possible for this engine to run at all under these conditions.

Is it surprising therefore that an engine, burdened with a lubricant of high viscosity, splutters and dies out on a cold morning even though the starter may have been turning it at 100 r.p.m. or more ? That it does start eventually is due to the fall in viscosity of the lubricant on the cylinder walls, owing partly to dilution with unburnt fuel and partly to the first few intermittent fires causing a rise in temperature.

Even when starting at quite normal air temperatures the internal friction is surprisingly high. If the indicated horse-power curve of an engine is plotted, as in Fig. 7, and the appropriate resistance horse-power subtracted for various values of oil viscosity, the results are startling. This particular engine, a six-cylinder engine of 2,400 cu. cm. capacity, gives an indicated horse-power of 73 at 3,500 r.p.m., but

it would not even reach this speed, running light, if all its lubricant remained at air temperature. When the working lubricant attains a temperature of about 45 deg. C., 23 b.h.p. is available at 3,500 r.p.m., and 48 b.h.p. at 75 deg. C. In starting from cold, all these stages have to be met and passed; though it should be noted that the cylinder walls, where most of the friction occurs, will have an appreciably higher temperature than the bearings while warming up.

Conclusion. The friction torque being expressed as $K\sqrt{NV}$, it is of interest to examine this expression with a view to possible reductions of

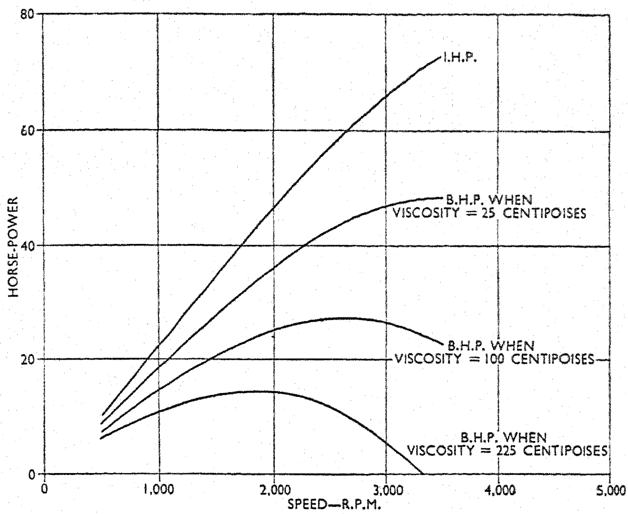

Fig. 7. Curves of Engine Horse-Power for Various Values of Oil Viscosity

2,400 cu. cm. six-cylinder engine.

frictional losses in practice. The biggest variant is V, the viscosity of the lubricant in centipoises. A heavy engine-oil at −5 deg. C. may give a value of $\sqrt{V}$ of 125, and a light winter oil at 100 deg. C., a value of 2·5. The bigger figure is encountered when cranking and during the critical period of the first few firing strokes; the smaller when the engine has been running an appreciable time. The effect of viscosity upon power output has been shown in Fig. 7, and since the losses are directly proportional to $\sqrt{V}$ it is obviously advantageous to use an oil whose viscosity under operating conditions is as low as is consistent with safety.

Much progress has been made during the last few years in the selection of lubricant for the motor car engine. Engines are being safely and economically run on oils of far lower viscosity than would have been considered practicable five years ago, and there are indications that this winter will see a further extensive move towards lighter oils and less engine resistance.

Originally it was considered that the magnitude of the factor K was entirely under the control of the engine designer. It is almost certainly dependent upon the dimensions and clearances of pistons and bearings, and also upon the type, number, size, and pressure of piston rings. It has already been shown that the swept area is an important consideration, and for an engine of given capacity this factor can vary within wide limits. Thus a 2-litre engine may have a swept area of 920 sq. cm. if it is a four-cylinder engine of short stroke, or 1,340 sq. cm. if it is an eight-cylinder engine of long stroke.

It now appears that K can also be influenced by a property of the lubricant other than its viscosity. The addition of oleic acid, whatever other effect it may have on cylinder or bearings, has been shown very noticeably to reduce the value of K below that obtained from straight blends of mineral oil.

The authors wish to make it quite clear that the subject matter of this paper has been obtained in the normal course of work in an engine research laboratory whose aim primarily has been to assist the motor industry in problems directly or indirectly affecting the electrical equipment. They wish to thank the directors of Messrs. Joseph Lucas, Ltd., for permission to publish results, and also Mr. C. G. Williams of the Research Laboratories of the Institution of Automobile Engineers for his helpful discussions.

THE ALTERATION OF LUBRICATING OIL DURING USE IN INTERNAL COMBUSTION ENGINES

By C. H. Barton, M.A.*

The evaluation of the performance of lubricating oils intended for internal combustion engines is admittedly a difficult matter. Actual running trials in engines are ultimately the only method of determining the relative behaviour of oils. Tests involving the use of engines are, however, costly both in money and in time and frequently involve difficulties in keeping experimental conditions constant. It is not surprising, therefore, that there has been a marked tendency to use chemical and physical tests for judging the relative performance of lubricating oils in service. The purpose of this paper is to consider briefly the significance of certain laboratory tests, especially those concerned with the oxidation and decomposition of lubricating oils in internal combustion engines.

The carbonaceous deposits found in the combustion spaces of petrol engines consist mainly of decomposition products of the lubricant. In compression-ignition engines the fuel or the combustion conditions may have an important influence on the formation of carbon deposits. The factors controlling the growth of carbon deposits in petrol engines are well known (Gruse 1933). The tendency of lubricating oils to leave carbonaceous deposits when decomposed at high temperatures under limited contact with air is compared by means of the so-called coke or carbon tests, of which the Conradson and Ramsbottom tests are the best known. These tests, however, do not completely reproduce the conditions under which the oil is cracked and decomposed in the engine. Although, therefore, coke tests may place a limited group of oils in the same order for carbonizing tendency as the results of engine experiments, a change in the engine or its condition of working may upset the correlation between laboratory coke tests and engine carbon deposits. Under constant running conditions the amount of carbon deposit tends to reach an equilibrium after about 20 hours (Bahlke and others 1931). If the quantity of carbon formed at this stage is sufficient to interfere with the proper working of the engine by, for example, promoting marked detonation, an improvement may sometimes be effected by changing, after cleaning the engine, to an oil of lower carbonizing tendency. In general, however, the modern engine is not very sensitive to differences in coke value between oils.

The conditions of use of lubricants in internal combustion engines

* The Asiatic Petroleum Company, Ltd.

involve high temperatures and oxidizing conditions. Oils differ according to their origin and refining treatment, both in manner and rate of oxidation; consequently, numerous methods have been proposed for evaluating the resistance of oils to oxidation in the laboratory. Two of the best-known tests of this nature will be considered, namely, that of the British Air Ministry and the Indiana oxidation test. The former, which is included in Air Ministry Specification D.T.D.109, involves oxidation of the oil for 12 hours at a temperature of 200 deg. C. by means of a stream of purified air which is passed through the oil at a prescribed rate. The changes in the oil brought about by this treatment are assessed quantitatively by measuring the viscosity before and after oxidation and by determining the increase in the "coke number", by the Ramsbottom test, due to the oxidation treatment. In the Indiana test (Barnard and colleagues 1934) the oil is maintained at 341 deg. F. (171·7 deg. C.) while a current of air is passed through at a specified rate. Samples of the oil are taken at intervals and examined for viscosity increase and presence of material insoluble in a light gasoline or petroleum ether. From these results the time of oxidation is found for the formation of 10 (and 100) milligrammes of so-called "sludge" per 10 grammes of oil. It was shown by Barnard and his colleagues that a number of lubricating oils of different origin and degree of refining fell in the same order of resistance to oxidation both in the Indiana test and in tests in a multicylinder air-cooled engine when the same criteria for degree of oxidation were applied to the used oils as to the oxidized oils after the Indiana test.

Although the Air Ministry oxidation test was originally intended for aircraft oils and the Indiana test was evolved for motor oils, it is not unfair to compare their results, since the conditions of oxidation and decomposition are essentially the same in the two classes of engine. In fact, the Air Ministry test is quite frequently applied to motor oils in this country with the object of comparing their resistance to oxidation. Table 1 summarizes the results found by the two tests, using four commercial lubricants.

It is evident that the two oxidation tests disagree as to the order in which they place the oils for stability towards oxidation. The Indiana test puts oil B first, followed by D and C, with A a very poor fourth. The Air Ministry test makes oil D the best, A second, and then C or B according to whether the coke increase or viscosity ratio is taken as the indication of stability. It is, however, rather exceptional to find such marked disagreement as that shown in the table, between the two oxidation tests, which generally agree in discriminating between oils of high and low resistance to oxidation.

The case of oils C and D is interesting, because these two oils differ

only in regard to the "finishing treatment" of the refining process. Both oxidation tests agree in assigning higher oxidation resistance to oil D, which also possesses greater stability towards light than oil C. No difference has, however, been found between the two oils in their behaviour in a petrol engine, even as regards the rate at which they form oxidation products.

Whatever their value for comparing the oxidation stability of mineral lubricants, it is becoming recognized that oxidation tests entirely fail to simulate the conditions of oxidation and decomposition in the engine. It may, for example, be shown that the oxidation products formed in

TABLE 1. COMPARISON OF OXIDATION TESTS

Oil	S.O.C. Indiana test. Time in hours taken to form per 10 gr. oil		Air Ministry oxidation test	
	10 mg. "sludge"	100 mg. "sludge"	Ratio of viscosities at 100 deg. F. after and before oxidation	Coke number increase
A. Aero-oil blended to meet Air Ministry specification . .	16	53	1·6	0·84
B. Solvent extracted distillate (viscosity 230 sec. Redwood No. 1 at 140 deg. F.) . . .	>200	>200	1·6	1·2
C. Highly refined "asphaltic base" distillate . .	58	80	1·9	1·14
D. Do. . . .	72	91	1·6	0·61

the laboratory consist principally of "asphaltenes", i.e. substances insoluble in petroleum ether but soluble in benzol. On the other hand, used oils contain a "sediment", insoluble in petroleum ether, which, in oils from petrol engines, consists roughly of 50 per cent asphaltenes, and, in oils from compression-ignition engines, of only about 10 per cent asphaltenes. The decomposition of lubricants in use is a very complex process (Pye 1934) which includes oxidation at temperatures usually below 100 deg. C. in the crankcase, accompanied by some cracking and carbonization of oil on the underside of the piston. In addition much of the oil which passes the pistons is decomposed by heat and leaves some of its products in the combustion

chamber. These products, together with condensed fuel and also materials formed by incomplete combustion of the fuel in compression-ignition engines, tend to collect in the oil on the cylinder wall and thence work down below the piston into the crankcase. The crankcase oil thus accumulates thermal decomposition products of the lubricant and even of the fuel formed under conditions quite different from those of the crankcase itself.

Although attention has been directed to the phenomenon from time to time (Thornycroft and Barton, 1930), adequate recognition has scarcely been given to the effect of decomposition products from the combustion chamber in contaminating the lubricating oil and forming sludge deposits. The question is, however, one of particular importance to the operators of high-speed compression-ignition engines on the road (Bouman 1937). This aspect of the deterioration of oils in service is obviously not covered by the usual oxidation tests.

As decomposition products are carried down to the crankcase by oil returning from the cylinders, it might be expected that an increase in the rate of circulation of the lubricant between the crankcase and combustion space would lead to enhanced oil deterioration. The measurement of the rate of circulation of the lubricant above and below the pistons has not yet been achieved, but an indirect indication of the rate of oil circulation is given by the rate of oil consumption which may be assumed to be a direct function of the rate at which oil passes the pistons in a given engine.

If the above picture of the events in internal combustion engines is correct, it follows that the rate of accumulation of decomposition and sludge-forming products in the crankcase oil should increase with the rate at which the lubricant passes the pistons and is consumed. It must not be inferred that all the decomposition products have their origin in the combustion chamber; some are undoubtedly formed in the crankcase, particularly when oil strikes the underside of the piston, but the destruction of oil at this point is generally much smaller in amount than that which takes place above the piston. Some evidence to this effect has been obtained by running a high-temperature engine with and without a "baffle" fixed inside the piston to prevent the oil stream from the big end impinging on the under side of the piston crown.

In engines which give a high consumption of oil in service, the more frequent addition of fresh make-up oil tends to mask the rapid deterioration due to the high oil consumption. In order to avoid the disturbing effect of adding fresh oil, experiments have been carried out in a single-cylinder bench engine without adding any oil after the start of the test.

The following results relate to 20-hour tests under identical conditions, except that in tests Nos. 2 and 3 the pressure in the oil system was increased by using a stronger spring in the relief valve of the oil pump to increase the rate of oil consumption.

TABLE 2. TWENTY-HOUR TESTS ON A SINGLE-CYLINDER ENGINE

Test numbers	1	2	3
Oil	A	A	C
Oil consumption, litres	0·5	1·3	1·7
Sediment, percentage by weight in used oil (insoluble in petroleum ether)	0·27	0·55	0·17
Oily sludge in used oil (by filtration), grammes per litre	3·5	7·4	4·0

It is evident that the used oil in test 2 is in a dirtier condition than that of test 1. The amount of oil put into the crankcase for each test was 4·5 litres. At the end of test 1, 4·0 litres remained; and at the end of test 2, 3·2 litres of "used" oil. The total amounts of oily sludge in the used oils are, therefore, 14 grammes and 23·7 grammes respectively. It follows, therefore, that the greater amount of sludge or sediment in the second test is not due to a concentration effect caused by the higher oil consumption. It will be noticed that the sludge and sediment

TABLE 3. RUNNING TESTS

Test number	4	5
Duration, hours	14	28
Total oil consumption, litres	1·25	1·2
Sediment, percentage by weight in used oil (insoluble in petroleum ether)	0·22	0·20
Oily sludge in used oil (by filtration), grammes per litre	2·1	1·7

contents of oil A are in a fairly constant ratio to one another. The results of test 3 have been added to show that when a different oil is introduced the relationship between sediment and sludge changes.

Another aspect of the same question has been studied by running tests at a high and a low rate of oil consumption on the same oil. Fresh oil was used for each test and the single-cylinder engine was

cleaned between the runs. Excepting for the adjustments to the relief valve of the oil system necessary to vary the oil pressure and rate of circulation, the conditions of running were constant throughout.

It is evident that the rate of deterioration of the oil in test 4 has been more than twice as rapid as that in test 5, thus showing that a reduction in the rate of oil consumption allows the oil to be kept in use for a longer period before it accumulates a specified content of oxidation and decomposition products.

It will be seen that in the results quoted above the sludge has been estimated by filtering the used oils, on the principle that sludge is essentially material insoluble in the oil. The filtration is done through paper, using suction. The sludge can also be removed by centrifugal treatment of the oil. The results of the two methods are in general proportional to one another. Both methods remove the sludge more completely from the oil than it separates in practice in the engine. The proportion of oil held in the sludge depends upon the conditions of filtration or centrifugal treatment. If it is considered preferable to eliminate the effect of the occluded oil on the results, the products of filtration or centrifugal operations may be extracted with petroleum ether before weighing. The "sludge sediment" results so obtained are in general quite different from those given by similar measurements carried out on the used oil before filtration or centrifugal treatment.

It is impossible in a short paper to deal completely with the effect of engine conditions on the deterioration of lubricating oil, but some indication has been given of the very marked effect of a high oil consumption in increasing the rate of deterioration of the lubricant. Anything, therefore, that can be done to reduce the rate of oil consumption in an engine not only effects a direct saving, but also prolongs the useful life of the oil.

References

Bahlke, Barnard, Eisinger, and FitzSimons. 1931 S.A.E. Journal, vol. 29, p. 215. See also Gruse, ibid.; Thornycroft and Barton. 1930 *Aircraft Engineering*, vol. 2, pp. 36, 109.

Barnard, Barnard, Rogers, Shoemaker, and Wilkin. 1934 S.A.E. Journal, vol. 34, p. 167.

Bouman. 1937 Second World Petroleum Congress, Paris.

Gruse. 1933 Am. Soc. Testing Materials, Symposium on Motor Lubricants, New York, 8th March; 1933 Proc. World Petroleum Congress, London, vol. 2, p. 441.

PYE, D. R. 1934 "The Internal Combustion Engine", Oxford University Press, vol. 2, chapter 5.

THORNYCROFT and BARTON. 1930 *Aircraft Engineering*, vol. 2, pp. 36, 109. See also LARSON. 1933 Am. Soc. Testing Materials, Symposium in New York, 8th March, p. 77; DIETRICH, ibid., p. 57. MAILLARD, ACKER, and RENGADE. 1936 *Annales des Combustibles Liquides*, Sept.–Oct., p. 945.

AERO-ENGINE DESIGN AND LUBRICATION

By E. L. Bass*

The attitude of mind exemplified by the expression "blame the oil" is now fortunately becoming extinct, owing to the better appreciation of the influence of certain details in engine design and operation upon lubrication. It is no longer the simple function of an oil to lubricate, it has also to cool and protect certain parts from corrosion under a wide range of temperatures and pressures. At the same time there must be such freedom from sludge and carbon deposits as to render possible overhaul periods of 500 to 750 hours. The subject has become still further complicated by the advent of high-octane fuels, enabling specific power outputs (and attendant heat losses) to be doubled within the last five years.

The pistons of aero-engines have, without doubt, proved the most difficult of all components to lubricate. They account for some 70 per cent of the total friction losses in the engine and may have to deal with about 30 per cent of the total waste heat. This is why aero-engine oils to-day are judged primarily by their ring-gumming properties. Experiments have shown that ring sticking can occur (as far as oil quality is concerned) either through the more or less rapid formation of some oxidized material in the ring grooves or through the gradual accumulation of sludge and carbon. Whilst the former can occur in a few hours, the latter takes effect only after some 200 to 300 hours' running. The two effects are due to entirely different properties of the oil and to different details in design. Given that the engine designer and metallurgist can provide pistons and cylinders not subject to distortion whilst running, lubrication of the piston skirt would cause little difficulty even at the elevated temperatures encountered. The necessity for using piston rings to provide the gas seal introduces a host of problems. There is, indeed, scarcely a single feature either in the principle of the piston ring or its operation which is not inimical to lubrication. Since they produce a very high proportion of the total piston friction they must be adequately lubricated. It is estimated † that 80 per cent of the heat dissipated by the pistons passes through the rings and the lands above the bottom ring. They are working with a sliding motion of non-uniform velocity and are subject to their maximum pressures and temperatures during a period of minimum velocity and actual reversal of motion. The oil supply, for the sake of economy

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* Chief Engineer, Shell Aviation Department.

† Pye, D. R. "The Internal Combustion Engine", Oxford University Press, vol. 2, p. 82.

in operation, is limited both on the friction surfaces of the rings and behind them in the grooves, where a more copious supply would assist in washing away oil sludge which might cause ring jamming and ultimately piston burning.

Both from the point of view of ring-gumming and efficient lubrication, the working temperatures of the pistons and cylinder walls must be kept within reasonable limits. The size of the cylinder and method of cooling are both of considerable importance in this connexion, but are usually determined by other considerations. The figures given in Table 1 indicate the variations in piston temperatures which have been observed on a variety of engines. Even the effect of turbulence must be taken into account, as is shown by the compression-ignition engine, for which piston temperatures as high as those in air-cooled aero-engines are recorded. In spite of the higher thermal efficiency of the compression-ignition engine it appears that the more rapid gas flow over the piston results in greater heat transfer. For the provision of extra strength and cooling area under the crown of the piston, "waffle-plate" and ribbed designs are now common. Even cooling fins behind the skirts have been found beneficial in some engines. Since such a high proportion of the heat flow from the piston passes through the piston rings, it is on the design and arrangement of these that success or failure depends. In general, two to three gas rings, and one or two scraper rings for oil control are used. The use of more rings for better heat distribution appears logical and warrants further investigation. Consideration of available space and increased friction will have to be weighed against the benefits accruing from reduced working temperatures. The tapered ring with the top, or top and bottom, faces having a 5 deg. taper with grooves similarly tapered, has proved a certain cure for ring gumming in a variety of aeroplane and high-speed compression-ignition engines. Whatever design of ring is employed, the side clearance in the grooves is of the greatest importance, the tendency to ring-gumming being almost inversely proportional to the clearance provided. Again, a compromise must be found as excessive clearance may advance oil consumption and cause "hammering" of the ring grooves. "Scuffing" or feather-edging of the rings is now practically eliminated by the use of suitable materials. At one time this trouble was thought to be due to lubricating oil, but only when relatively volatile oils are used is this true. In any case, such oils are now considered unsuitable for the lubrication of high-output engines for other reasons.

Given an engine free from oil leaks, practically all the lubricating oil consumed is that passing the pistons into the combustion chamber. The speed of rotation of the engine is the chief factor which controls

TABLE 1. VARIATIONS IN PISTON TEMPERATURES *

Type of engine	Capacity per cylinder, litres	Cooling medium	Temperature, deg. C.	Brake mean effective pressure, lb. per sq. in.	Speed, r.p.m.	Horse-power per sq. cm.	Horse-power per litre	Piston temperature, top land, deg. C.
(1) Stationary; petrol	0·75	Liquid	190	66	1,100	0·061	5·6	300
(2) Stationary; compression-ignition	0·50	Liquid	190	72	1,200	0·073	6·7	320
(3) Stationary; petrol	2·50	Liquid	190	81	1,000	0·099	6·2	250–290
(4) Automobile 2-stroke; petrol	0·35	Liquid	85	62 two-stroke =124	3,000	0·218	28·7	250–300
(5) Automobile 4-stroke; petrol	0·32	Liquid	75	41	2,500	0·073	8·1	160
(6) Aero 4-stroke; petrol	3·00	Air	—	170	2,500	0·540	32·6	340–350

* Piston temperatures measured by the fusible plug method.

this, apart from the design of the piston and ring assembly. The full-skirted piston with one scraper at the bottom—below the gudgeon pin—and one immediately below the gas rings finds most favour to-day. A wide variety of scraper rings is now available, but it would seem that their location, clearances, and the oil relief holes in the grooves are of rather greater importance than the relatively small differences in scraper ring sections.

Even when satisfactory designs of piston assemblies have been developed, operating conditions can have a profound effect on results obtained, neglecting, of course, gross misuse of an engine by overloading, etc. For example, slight detonation accelerates ring gumming to a remarkable extent. Experiments on small air-cooled engines have shown that the occurrence of slight detonation will reduce the ring-sticking temperature of a given oil by as much as 20 deg. C. The air-fuel ratio employed also has a profound influence on ring sticking. The weak mixtures employed under cruising conditions on modern airline and long-range aircraft considerably increase the tendency to oxidation of the lubricating oil. This effect is, of course, distinct from that of the higher temperatures occurring with mixtures coming in the "theoretically correct" region. Therefore the complete testing of either lubricating oil or piston and cylinder assembly must involve experiments with both detonating and weak mixtures.

About the design of the cylinder there is little to be said, except to mention the necessity for adequate cooling and freedom from distortion. In this connexion, the cooling of the bottom of the cylinder barrel must not be overlooked, lubrication difficulties having been encountered on engines deficient in this respect, although the cylinder head and top of the barrel were adequately cooled. Generally speaking, wear on cylinder barrels has not been a very serious problem with aero-engines in the past, a fact which would appear to confirm the corrosion theory of wear, as the warming-up period of air-cooled engines is very short and extremely little running is done when the cylinders are not at their normal operating temperatures. Some of the latest types of high-output engines, however, have been found to suffer from excessive cylinder wear. The solution to this problem appears to lie almost entirely in the selection of suitable materials and the treatment of the cylinders by processes such as nitriding. It has been suggested that this trouble can be overcome by the use of compounded and fatty oils known to have greater "oiliness" than the straight mineral oils. Generally speaking the difference between compounded and straight mineral oils in this respect is so slight that the former do not provide an adequate factor of safety, even if they are successful in preventing excessive wear on one particular engine.

Whilst modern engine designs provide for the automatic lubrication of practically all engine auxiliaries, it is not possible to lubricate the rocker gear of some valve layouts used in radial engines by this method. Hydraulic tappets offer means of overcoming the mechanical difficulties involved in this connexion. The problem of valve stem lubrication both when running and standing (for the avoidance of cold corrosion) is also becoming more difficult to solve in recent designs, due to the use of the fully salt-cooled exhaust valve. With this, the heat flow through the stem to the guide is greatly increased and the selection of suitable materials for these components becomes of major importance. All difficulties connected with valve gear lubrication, however, are eliminated with the sleeve-valve type, of which some most promising designs are already in production.* In these, there has been considerable ingenuity displayed in detail design to overcome the chief bugbears of the sleeve-valve system, namely, high oil consumption and gumming of head rings. Actually, for the lubrication of the piston and cylinder, the sleeve-valve engine offers considerable advantages since there is no direct reversal of motion of the sleeve in relation to either piston or cylinder.

Sludge formation, due to contamination of the oil by products of combustion, and by oxidation of the oil, is objectionable especially in civil aero-engines, where length of time between overhauls influences the economics of operation. Here it is interesting to observe that experience shows that the minimum of sludging occurs in efficient engines running with low oil consumption. It is admitted that lubricating oils differ greatly in this respect, and the properties controlling this characteristic are well known. Nevertheless, even the most suitable oils available can produce undesirable deposits in certain engine parts after long periods of running, if the design is such that a relatively rapid oil flow is not maintained in the places where sludge deposits or settles out. Oil sludge will accumulate rapidly in "dead ends" subjected to centrifugal force and although this may be used as a form of sludge separator (e.g. "flywheel" type filters), sludge accumulation may become an undesirable feature in reduction gear shafts, blower clutches, etc. Although a certain amount of sludge may collect even when adequate oil flow is provided, it is practically unknown for a complete stoppage of oil holes to occur under such circumstances. The rate of flow increases as the oil channel decreases in size, so that a condition of equilibrium is reached and then no further deposition of sludge takes place.

The inclusion of filters in engine lubrication systems is now general practice. These are only capable of the removal of extraneous matter

* *Bristol Review* (Engine Issue), 1936, No. 9, May.

and carbon from the lubricating oil which is necessary for the protection of bearings. Many improved designs of filters are now available, and are being fitted as standard to aero-engines. Some of these filters are of the fabric type with renewable elements, and others are of the "Autoklean" type with either automatic (Wright Aeronautical Corporation) or manual control.

In military aviation the time taken between starting-up and take-off is of particular importance. The factor controlling this time is the warming-up period for the oil, and various devices have been developed to enable adequate oil circulation to be obtained immediately on starting. The Bristol and United States Army methods * are the best known of these. A further development is the predilution of the oil with gasoline, which has been used successfully in the United States.

Recent developments enabling higher engine outputs to be obtained have added considerably to the design problems of big-end bearings, and have introduced the need for copper-lead and cadmium-base bearing metals. Mechanically these materials are superior to the whitemetal bearings generally used in the past. The same developments necessitating these new bearing metals have also demanded new qualities of oils of improved heat-resisting properties. The special refining processes necessary for the production of these oils appear to remove certain protective qualities from them. In some cases, when such oils have been used in conjunction with copper-lead and cadmium-base bearings, severe corrosion of the bearings has been observed. Whilst this is largely a question of bearing temperatures (corrosion not appearing below about 130 deg. C.), it seems impracticable to obtain working temperatures low enough to avoid corrosion. Development of new bearing materials and methods of manufacture of them promise to overcome these difficulties. At the same time the use of special "dopes" in the lubricating oil has proved efficacious in some cases. Both aspects of the problem are now under investigation.

In a brief paper such as this, reference to many problems having a profound though secondary effect on lubrication has been omitted. As examples may be quoted the cooling and baffling of air-cooled engines, the use of liquids of high boiling point (e.g. ethylene glycol) as the cooling medium in liquid-cooled engines, oil coolers and their installation, etc.

In conclusion, it is of interest to observe the influence of present trends in engine design upon lubricating oil development. Higher specific powers resulting in increased temperatures and heat flows demand oils of improved heat resistance. This property of an oil

* Worth, W., S.A.E. Journal, 1937, July, "Lubrication and Cooling Problems of Aircraft Engines".

can only be judged by engine tests, for which single-cylinder aero-engines have been found most satisfactory. The finding of a technique for such tests has only been achieved by few, and has demanded considerable expenditure of money and patience.

Even the production of oils satisfactory in respect of heat resistance by no means satisfies all requirements of modern high-output engines. The desirability of using compounded oils (either with fatty oils or special dopes) is at present receiving the closest investigation. Their advantages in respect of improved oiliness must be weighed against their greater sensitivity to oxidation, although there are now indications that this disadvantage may be avoided. There is still one further development in engines which may involve modifications to the present lubricating technique. With rapidly increasing power outputs, the question of the tooth loading of reduction gears becomes a serious item. The automobile industry has shown how this difficulty may be overcome by the use of extreme-pressure lubricants. Unfortunately these oils are quite unsuitable for engine lubrication. If extreme-pressure lubricants become a necessity in aero-engines it will clearly involve isolation of the gear lubrication system from that of the engine and auxiliaries.

The question of the sludging properties of aero-engine oils is of great importance and introduces the subject of compounding, as the solvent effect or capacity for keeping the sludge in suspension with certain of these oils is well known. The development of aero-engine oils appears generally to have lagged behind that of their fuels. The chief reasons for this are the great difficulty involved in testing oils and the extremely contradictory results obtained both experimentally and in service. Such observations only show the extent to which the various mechanical factors mentioned in this paper mask the differences between lubricating oils. The solution to the numerous problems involved, sometimes seeming hopeless to those engaged upon them, can only be obtained by the expenditure of almost unlimited time and money upon engine testing. Co-operation between oil producer and engine manufacturers becomes essential for this.

OIL RECLAMATION AND THE USE OF RECLAIMED OIL

By Albert Beale, A.M.I.Mech.E.*

Used lubricating oil from an internal combustion engine, which has become so dirty in use as to need replacement, may be reclaimed in a condition equal to new in lubricating value so that it can be re-used for its original purpose. This is a useful and important fact, but one may go further and state that lubricating oil used in an internal combustion engine may, by filtration, be prevented from becoming appreciably dirty, and this is even more useful and important.

It is common practice to fit in the lubricating system strainers which pass the full flow of oil and remove the coarser impurities as they arise. Such a procedure is necessary, but it is not sufficient. It is not disputed that even the best strainer does not remove the finer impurities from the lubricating oil, and these, therefore, accumulate to an increasingly objectionable extent. With an internal combustion engine, and especially an engine of the compression-ignition type, the necessity for further action beyond the incorporation of a strainer in the engine oil system, quickly becomes apparent, and one of the following three courses must be taken:—

(1) The oil may periodically be drained from the system and replaced by new oil.

(2) The oil may periodically be drained from the system, efficiently filtered, and re-used.

(3) The oil actually in the system may be kept constantly free from accumulations of fine impurities by an efficient filter working on the bypass principle.

The most obvious objection to the first course is that of expense. There is, however, a much more serious objection, namely, that considerations of expense or difficulties of disposal discourage the draining of the oil from the system as often as is really necessary and there is a consequent amount of engine wear and tear which might well have been obviated.

With regard to the second course, namely, draining, filtering, and re-using, compared with the first, this enables a saving in lubricating expense amounting to from 75 to 90 per cent according to engine operating conditions. Frequent draining of the oil from the sumps is encouraged and the expense of engine wear and maintenance is thus

* Stream-Line Filters, Ltd.

reduced. The procedure is not ideal, however, for instead of the engine running continuously with clean oil, it is operated with oil which is allowed to become dirty to a certain degree before draining for filtration.

The third course, namely, keeping the oil reasonably clean at all times without draining from the sump, is ideal from the point of view of correct engine operation. The use of an efficient bypass filter to keep the oil in circulation continually clean is usually, but not invariably, commercially practicable. In road transport vehicles an efficient filter for use on each vehicle is often larger than can conveniently be accommodated, and it is also bound to be relatively expensive; whilst reliance cannot always be placed upon the proper maintenance of filters on the individual engines of a large fleet. In such cases periodic sump drainage and filtration, the second of the methods mentioned above, may often be the best practical arrangement even though bypass filtration is better in theory.

Because of the technical superiority of bypass filtration this method is taken first in the brief consideration of methods employed which follows.

Types of Bypass Filter

Practical types of filter for use in bypass filtration are really limited to those based on two main principles: centrifugal separation and "Stream-Line" filtration. In centrifugal separators the oil is fed through a specially designed bowl rotated at speeds varying from 8,000 to 16,000 r.p.m., and as a result of the intense settling action which takes place in the comparatively thin layer of oil in the separator bowl, all but the fine solid particles can be separated out. The oil can be made to flow through the separator at any desired speed, but the efficiency with which the impurities are removed will naturally vary with the speed, so that for any given plant there is a critical rate of flow which results in a given time in the maximum removal of solids.

"Stream-Line" filters work on the basis of edge filtration carried to a very fine degree. The oil flows between a very large number of paper disks piled in columns under end compression in such a way that the oil under a suitable pressure difference passes from the outside to the inside of the columns of filtering disks, whilst the solid impurities do not penetrate to the least extent beyond the paper edges. This process enables even the very finest impurities to be eliminated, though necessarily the filtration rate obtainable from a plant of reasonable size is smaller than in the case of less efficient purifiers.

With a filter of 100 per cent efficiency in operation on the bypass

principle the dirt content of the oil in the engine system can never exceed a certain fixed maximum value, and with a filter of quite moderate size this maximum value may be made so small that the engine is always working in something approximating to new oil conditions. The question has been investigated mathematically in a paper by the author (1936 *), many practical details being given.

Although with a centrifugal separator working on bypass there is bound to be some accumulation of the very finest impurities, large particles are eliminated, and with the constant addition of a certain amount of new oil for make-up purposes the oil even in this case is prevented from becoming contaminated to the extent which would occur without bypass purification.

Without going further into details concerning the relative merits of different types of attachment filters for bypass treatment of oils it may be said that there is already an enormous body of experience which establishes conclusively that the oil may be kept in satisfactory service in the engine indefinitely whenever an efficient system is employed.

Reference is made later, when considering the results obtained by means of batch filtration, to the significance of the observed result that bypass filtration properly undertaken keeps the oil in good condition for an indefinite period. For the moment the fact need merely be stated, and coupled with the remark that the condition of the oil remains satisfactory not merely because of the absence of impurities but also because the growth of acidity, saponification value, and indications of damage to the oil by oxidation are negligible.

Types of Batch Filter

Coming to the question of batch filtration, i.e. periodic sump drainage and re-use of the oil after treatment, it will be clear that, when dealing with the oil quite apart from the engine, the way is opened up to a number of methods of treatment which could not conceivably be employed on the bypass principle. The methods available are roughly classified below:—

(*a*) *Pure Filters*. Under this heading there are two types, the pad filter and the "Stream-Line" edge filter. Of these the latter has already been described and it can be equally well applied in the bypass or batch system. Pad filters are made in various forms, all of which embody the principle that the impurities to be filtered out are removed by becoming enmeshed in a porous filtering medium. With the edge filter, in which all the solid impurities are removed at the paper edges,

* Diesel Engine Users' Association, 1936, December.

the deposit of impurities can be removed periodically in a very simple way by reversing the direction of flow with compressed air, but in a pad type filter cleaning of the pads is a virtual impossibility, and the process to be adopted, therefore, is the renewal of the pad. For internal combustion engines oil pad filters have been very little applied.

(*b*) *Settling Systems.* Cleaning oil by gravity settling, with the assistance of heat and in some cases water washing, can be quite successful apart from the considerations that the equipment is very large and cumbersome and that there is a considerable loss of oil with the sludge. Because of these facts, gravity settling systems are now very little used, but centrifugal separators form a practical type of equipment.

(*c*) *Chemical Cleaning Plants.* These are in effect settling systems in which gravity settlement is assisted by the addition of chemicals for coagulating purposes. The procedure is not to be recommended because of the danger that in unskilled hands chemical impurities may be left in the oil, whilst there is also a tendency for the percentage loss of oil with the resulting sludge to be very large. Combinations of chemical cleaning with centrifugal separation are, however, used with some degree of satisfaction.

(*d*) *Central Re-refining Plants.* It would, of course, be possible to take a used oil and, regarding it as if it were a crude oil, to subject it to very much the same processes as are employed in the original production of the lubricant from the crude so as to produce virtually a new lubricating oil. Re-refining plants are used to some extent on the Continent, although for reasons which will appear they have not been very much adopted elsewhere. The methods employed usually involve distillation and treatment by means of activated earths, and there are objections in the high initial cost of plant, the necessity for skilled operators, and high operating expense, as well as high loss of oil.

The supposed advantage of the central re-refining plant is that it takes oil in however bad a condition and not merely eliminates solid impurities but also corrects acidity. In order to justify the expense of the plant it is necessary to handle large quantities of oil and this means collecting dirty oil from a large variety of sources. In effect, the complexities of the re-refining plant are largely made necessary because of the mixture of oils being treated, and the mixture would never arise if the oils were treated by their actual users.

General Conclusions

Reference may now be made again to the results of bypass filtration. Engines are common in which the oil has not been changed in the sump during four or five years, and there is a large number of fully authenti-

cated instances where the oil has not been changed in ten years, and yet the lubrication of the engine is at least as satisfactory as at the outset, if not better. This demonstrates that bypass purification, in addition to keeping down mechanical impurities, actually prevents the development of acidity and other indications of changing oil structure. Yet if oil is used for a prolonged period in an engine without a bypass filter, or if the removal of the oil for treatment by a batch filter is unduly delayed, the oil may reach a condition in which plain mechanical filtration is not enough and the re-refining process becomes necessary. It would seem, therefore, that the presence of fine solid impurities and traces of moisture has a catalytic action, inducing the development of acidity and the processes of oxidation. In the absence of these fine impurities, which can be ensured by bypass filtration or frequent batch filtration, the oil remains in good condition and re-refining processes never become necessary. The oil user may well be advised for reasons of oil economy and engine maintenance alike, to prevent his oil from becoming dirty rather than to seek remedies when it has.

The title of this paper may perhaps be thought to suggest that the matter to be dealt with should concern exhaustive processes for the reconditioning of oil which has become utterly unsuitable for use. It is certainly a characteristic of human nature that there is more obvious enthusiasm over the reconditioning of one gallon of thoroughly dirty oil than over the constant cleanliness of thousands of gallons of oil which are regularly cared for and need no reconditioning. However, putting aside the inclination for spectacular achievements, it is suggested that accumulating experience must lead to the full appreciation of the advantages of constant bypass filtration or frequent batch filtration, which enable engines to run always on good oil, and at the same time enable the filtration process to be carried out by simple mechanical means which the engine user himself may control with perfect confidence and safety. It is with the importance of this point in mind that the author has produced the conjunction between the title of the paper and the present subject-matter.

LUBRICATION OF PISTON RINGS

By C. A. Bouman*

With most aircraft engines and high-speed Diesel engines the piston rings present difficulties to both engine designer and lubricating oil manufacturer. Although considerable improvements in design and in lubricating oil have already been obtained, much still remains to be done.

One question is whether lubrication between the rings and the cylinder wall is of the fluid or of the boundary type. The answers given to this question by various investigators show important differences, some stating that the lubrication between the rings and the cylinder wall is of the fluid type, whilst others claim boundary lubricating conditions (cf. Ricardo 1922; Stanton 1924; Sparrow and Thorne 1927; Marshall and Barton 1927; Hawkes and Hardy 1936; Taylor 1936). In the author's opinion there is no doubt that *both* types of lubrication, with gradual transitions from one type to the other, prevail between the piston rings and the cylinder wall.

Lubrication between a piston and its cylinder wall will tend towards the boundary type for piston positions near the dead centres, where the piston speed is low; for intermediate positions, where the piston speed is higher, the tendency will be towards fluid lubrication. Whether, to what extent, and where these tendencies will be realized depends largely on engine conditions. Amongst these, the amount of lubricant supplied to the cylinder wall is of the utmost importance, as lack of lubricant extends the region of boundary lubrication and increases cylinder and ring wear. The explanation is simple: the reciprocating motion of the piston tends to carry away the oil film from the cylinder wall so that, unless sufficient oil is supplied to the cylinder during each stroke, at some points of the cylinder wall the fluid lubrication will soon become boundary lubrication and greater cylinder wear and ring wear will result. In other words (Sparrow and Scherger 1936), it may be said that cylinder wear is found *where* lubrication is inadequate and it occurs *because* the lubrication is inadequate.

Of course, some restriction to this general statement becomes necessary when wear due to abrasion and corrosion is considered. The lubricating conditions still remain of great importance for those regions of the cylinder wall which are liable to abrasion and corrosion. Thus Williams (1933) found that an increase in the amount of lubricating oil fed to the cylinder of a gasolene engine resulted in a considerable

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decrease of the wear due to corrosion. In a remarkable paper Taub (1937) pointed out that European motor car engines in general show higher bore wear than American engines. According to Taub, the reason for this difference is inadequate oil supply to the cylinder walls of the engines of European make. It is well known that with two-stroke engines there is more danger of insufficient lubrication than with four-stroke engines, not only because in the former the piston temperatures are higher, but also because the scavenging and exhaust ports tend to scrape off the oil film from the piston skirt.

In general, the lower the oil viscosity, the higher the load on the piston rings (gas pressure behind the rings) and the lower the speed the more easily the lubricating oil will be squeezed out from between the cylinder wall and the rings. Although an adequate supply of lubricating oil between the cylinder wall and piston is necessary in order to secure fluid lubrication as far as possible, it should be kept in mind that it is impossible to reduce cylinder and piston ring wear below a certain limit, even with a large excess of lubricating oil. There will always be some boundary lubrication in the vicinity of the dead centres, owing to the very low velocity of the piston at those parts of the stroke.

In most cases the use of well-known oiliness additions, e.g. stearic acid, to the lubricating oil does not cause a measurable improvement in piston friction losses and piston ring wear (even in starting tests). This negative result may serve to indicate that the amount of lubricating oil fed to the cylinder wall is usually quite sufficient.

Ring Sticking. Another important aspect of the lubrication of piston rings is the phenomenon of ring sticking, which occurs in many internal combustion engines and may cause serious trouble. In general, ring sticking is caused by deposits in the ring grooves. These deposits may result from deterioration and carbonization of the lubricating oil in these grooves or from the accumulation of contamination products formed elsewhere.

A distinction has been drawn (Bouman 1937) between ring sticking at high engine temperatures and ring sticking at low engine temperatures. Low-temperature ring sticking is usually caused by oxygenated lacquer products formed during incomplete combustion of the fuel at low loads. These lacquer products are deposited on the cylinder wall, whence they are partly carried by the piston rings into the ring grooves (Boerlage 1930, Bouman 1933). These cases of ring sticking occur especially with some types of hot-bulb engines and with air-injection Diesel engines. High-temperature ring sticking, the more frequent type, may occur in any petrol or Diesel engine running at high piston

temperatures. The higher the temperatures in the engine the greater the tendency to this type of ring sticking.

With gasoline engines as well as Diesel engines the influence of the combustion process on the tendency to ring sticking is very important. For gasoline engines it is especially the anti-knock value of the fuel that is important, as detonation causes an increase of the piston temperatures. For Diesel engines sooty combustion of the fuel at high load will greatly increase the tendency to ring sticking, because much soot is caught in the lubricating oil film on the cylinder wall, from which it reaches the piston ring grooves. Moreover, sooty combustion coincides mostly with after-burning, which causes higher piston temperatures and, consequently, greater tendency to ring sticking. In addition to the piston and cylinder temperatures many other engine factors may influence the tendency to ring sticking; these factors will not be discussed further in this paper.

The deposits in the ring grooves which cause the rings to stick consist mainly of the following products:—

(1) Carbonaceous deposits, resulting from deterioration and carbonization of lubricating oil in the ring grooves, on the lands, the top land, and on the rim of the piston crown.
(2) Soot deposits from combustion.
(3) Other products of contamination, such as wear products from the engine, ash from the fuel, and dust from the atmosphere.

According to the operating conditions of the engine, the consistency and the composition of the deposits in the ring grooves may vary widely. Analysis of these deposits shows that they mainly contain products which can be separated according to their solubility in various solvents. The following classification is used at Delft:—

(1) Oil	Soluble in benzine (60–80 deg.).
(2) Lacquer . . .	Insoluble in benzine (60–80 deg.); slightly soluble in benzol; soluble in alcohol.
(3) Asphaltenes . .	Insoluble in benzine (60–80 deg.) and alcohol; soluble in benzol.
(4) Carbon, soot, and ash .	Insoluble in benzine (60–80 deg.), alcohol, and benzol.

When using mineral lubricating oils the percentage composition of the deposits in the ring grooves is mostly as follows:—

	Per cent
Lacquer	1–5
Asphaltenes	1–5
Carbon and soot	40–80
Ash	2–5, in special cases even 10.
Oil	from a few units up to a maximum of 40–50.

In Diesel engines the deposits in the ring grooves generally contain much more soot than in gasolene engines. The percentage of ash in the deposits is dependent on the ash content of the fuel. Von Philippovich (1937 a, b) points out the high oxygen content of the carbon deposits in the ring grooves and concludes that carbonization of oil in the ring grooves is not the only process which causes sticking of the rings.

When the engine conditions are changed the intensities of the various processes (oxidation, polymerization, carbonization, etc.) to which the lubricating oil in the ring grooves is exposed will change also. In the author's opinion these changes in intensity may be different for different lubricating oils: two oils which, under certain engine conditions, show a certain difference in their tendency to cause ring sticking, may be classified differently with respect to each other when tested under other engine conditions. However, the classification of various lubricating oils as determined in small oil-testing engines (both petrol and Diesel engines) generally agrees pretty well with the classification of the same oils in practice (Bouman 1937, von Philippovich 1937a, b).

Improvement of the lubricating oil with respect to its tendency to cause ring sticking may be obtained either by the addition of some anti-ring-sticking material to the oil or by the improvement of the oil itself. Really effective anti-ring-sticking "dopes" may or may not reduce the amount of carbon deposit formed in the ring grooves. In the latter case the anti-ring-sticking action of the dope is based exclusively upon some other effect in delaying the sticking of the rings when carbon deposits from the oil have already formed. Of both types of dopes some examples are known which show an important anti-ring-sticking effect.

To improve the oil, it must be made more resistant to evaporation and deterioration under the conditions existing in the ring grooves. When the residue from the evaporation and deterioration of a certain oil is only small the tendency to ring sticking will be less than with an oil having the same resistance against evaporation and deterioration but leaving a greater residue in the ring grooves. A low Conradson carbon number of the oil will be of some help, but an oil with a high Conradson carbon value, but with a low volatility, may show less tendency to ring sticking than an oil with a lower Conradson carbon value but with a greater volatility.

Figs. 1 and 2 show the results of tests obtained with a single-cylinder horizontal gasolene engine with cooling by the evaporation of ethylene glycol, running with three different oils A, B, and C. The principal data for these oils are given in Table 1.

As the fresh oils did not contain abnormal additions of high volatility

(gasoline or other low-boiling components) the flash point may be considered as a rough indication of the volatility of the oil.

During these tests practically no detonation occurred. The temperature of the cooling liquid was about 190 deg. C., this being

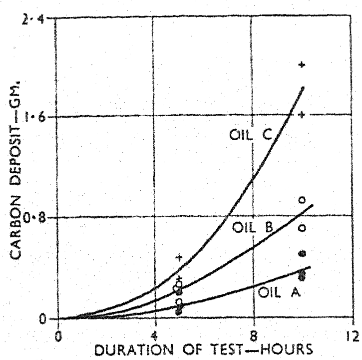

Fig. 1. Carbon Deposits in Ring Grooves and on Lands

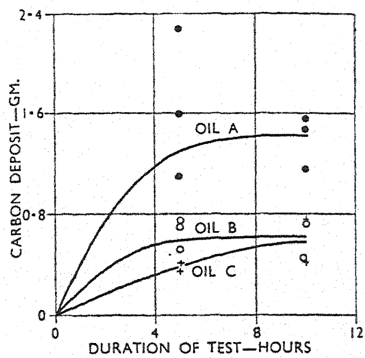

Fig. 2. Carbon Deposits on the Piston Crown

the boiling point of the ethylene glycol. The piston temperature near the ring grooves was about 260–280 deg. C., and in the centre of the piston crown about 310 deg. C. These temperatures were estimated with the aid of small pins of low melting points; by successively using pins of different melting points in the same hole, the limits could be found for the local temperature.

TABLE 1. PRINCIPAL DATA FOR FRESH OILS

Lubricating oil	A	B	C
Viscosity at 100 deg. F., centistokes	1,200	230	295
,, at 50 deg. C., ,,	506	123	140
,, at 210 deg. F., ,,	53	21	19
Kinematic viscosity index	95	110	80
Closed flash point (Pensky-Martens), deg. C.	290	250	210
Open ,, ,, (,,), deg. C.	330	290	230
Conradson carbon test, per cent	3·0	0·5	0·2

From Fig. 1 it appears that oil A (low volatility) gave the best results as regards carbon formation in the ring grooves and on the lands; this oil showed the least tendency to evaporate and carbonize under the temperature conditions prevailing near the ring grooves. As regards carbon deposits on the piston crown, however, the rating of the three

oils was just the opposite, the oil A with the high Conradson carbon value now causing more deposits than the other two oils (Fig. 2).

It would be desirable to ascertain whether oils of very high viscosity and very low volatility (like oil A) show advantages over normal commercial oils when used under practical conditions in engines with high operating temperatures, such as aviation engines. Obviously, the practical use of such oils would be restricted to those cases where preheating of engine and oil before starting can be accomplished and where a sufficiently high oil temperature during operation can be maintained.

REFERENCES

BOERLAGE, G. D. 1930 Jl. Inst. Petroleum Technologists, Oct, p. 678, "The Cause of Gummy Pistons".

BOUMAN, C. A. 1933 World Petroleum Congress, London, "Lacquer Formation in Diesel and Semi-Diesel Engines".

1937 Second World Petroleum Congress, Paris, "Gommage des segments de piston dans les moteurs à combustion interne".

HAWKES, C. J., and HARDY, G. F. 1936 Trans. North East Coast Inst. Eng. and Shipbuilders, vol. 52, p. 143, "Friction of Piston Rings".

MARSHALL, A. G., and BARTON, C. H. 1927 Jl. Soc. Chemical Industry, vol. 46, p. 130T, "Lubricating Oils: Laboratory Tests in Relation to Practical Results".

RICARDO, H. R. 1922 *Automobile Engineer*, vol. 12, p. 304, "Some Recent Research Work on the Internal Combustion Engine".

SPARROW, S. W., and THORNE, M. A. 1927 National Advisory Committee for Aeronautics, Report No. 262, part 2, "Friction of Aviation Engines".

SPARROW, S. W., and SCHERGER, T. A. 1936 S.A.E. Journal, vol. 38, p. 117, "Cylinder Wear, Where and Why".

STANTON, T. E. 1924 Aeronautical Research Committee, Report No. 931, "Friction of Pistons and Piston Rings".

TAUB, A. 1937 *Automobile Engineer*, vol. 27, p. 134, "Cylinder Bore Wear".

TAYLOR, M. P. 1936 S.A.E. Journal, vol. 38, p. 200, "The Effect of Gas Pressure on Piston Friction".

VON PHILIPPOVICH, A. 1937a *Luftfahrtforschung*, vol. 14, p. 254, "Über die Beständigkeit von Flugmotorenöl und ihre Prüfung".

1937b Second World Petroleum Congress, "Die Veränderung von Flugmotorenöl im Betrieb und ihre Prüfung".

WILLIAMS, C. G. 1933 *Automobile Engineer*, vol. 23, p. 259, "Cylinder Wear".

OIL PURIFICATION

By A. G. Cahill*

Apart from the selection of a suitable quality of oil for a particular duty, the most important consideration is the maintenance of its lubricating qualities. It is frequently stated that the lubricating value of an oil deteriorates during service. This is not strictly true. A certain amount of chemical change sometimes takes place, but providing the oil is suitable for the duty, this change is usually only slight and rarely reduces the lubricating value to any great extent. The oil does become contaminated with dirt, metallic particles, water, and sludge, and the lubricating value of this mixture is naturally less than that of new oil. Provided that this adventitious matter can be removed, the lubricating value will be equal to that of new oil.

Formerly, the usual method of conveying oil to bearings was by the drip or wick system, where small quantities were fed to the bearings at a rate adjusted by the position of the wick, which also acted as a filter. The oil passed through the bearings, carrying with it any impurities it had collected during its passage, and was replaced by new oil.

With the demand for high power with limitation of size and weight, drip lubrication became inadequate to handle the increased bearing speeds and loads, and either full forced, or a combination of forced and splash lubrication on the circulating system is now practically universal. By flushing large quantities of oil through the bearings, adequate lubrication is provided, and at the same time the excess heat generated is conducted away by the oil. As it is impossible to arrange for a continual supply of new oil in the quantity required, the oil is drained to a sump and recirculated. This results in contamination by dust, sludge, metallic particles, water, acid, etc. These can be removed by various means and to varying degrees, but the engineer is mainly concerned with the simplest and cheapest method of removing sufficient of these harmful additions to maintain the oil as an effective lubricant. Obviously, the ideal system would remove every particle of foreign matter from the oil continuously, by a plant with a capacity equal to the rate of circulation, but this is ruled out on the score of cost, and only a certain proportion of the quantity in the system is treated. Assuming that the amount of contamination is directly proportional to the number of hours run, there are two methods by which the con-

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tamination can be kept below the danger point. These are generally known as the " batch " and the " bypass " systems.

Systems. With the batch system (Fig. 1) the oil is used for a certain period, either completely or partly drained off, and replaced by clean

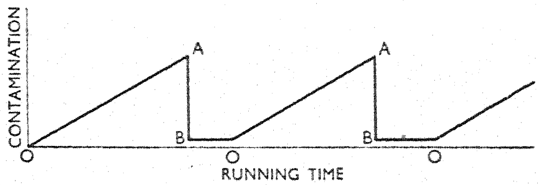

Fig. 1. Batch Purification in Service

A Engine stopped. Dirty oil drained out.
B Fill with clean oil.
O Start.

oil. The dirty oil is then purified as a "batch" and stored for further use. Even though the sump is filled with clean oil, a certain amount of contamination takes place before the oil reaches the bearings. This is due to the portion of the dirty oil left in the system, which cannot be drained off completely.

In the "bypass" or "continuous" system (Fig. 2), a proportion of the oil is constantly withdrawn from the circuit, cleaned, and at once

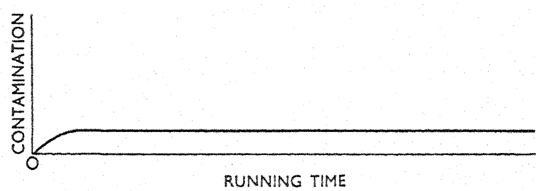

Fig. 2. Continuous Purification in Service

O Start engine and purifier.

returned. The rate of treatment depends on the type and operating conditions of the engine and will be dealt with later.

Comparison of Systems. The principal advantage of the batch method lies in lower lubrication costs due to the provision of reclaimed oil, instead of new oil, for refilling. This, in turn, allows for more frequent changes. The disadvantage, as compared with the bypass or continuous system is that there must be some point on the line OA (Fig. 1)

at which adventitious matter will begin to affect the lubricating properties of the oil. After this point the wear and the temperature of bearings will increase until the oil is changed. This normally involves stopping the plant, so that the frequency of the operation is usually governed by convenience of shutting down, rather than by the state of the oil. For these reasons the batch system is only recommended where it is impossible to employ the bypass method, or where foreign matter is present which cannot be removed without special treatment.

With the bypass system only a small amount of foreign matter can be present in the oil at any time. The high rate of circulation employed for bearing oil is normally sufficient to keep impurities in suspension, and as the oil is treated whilst circulating, the impurities are fed to the purifier and removed. In the batch system, where the circulation is stopped while the oil is drained down, a certain amount of foreign matter will be deposited in the piping, and in pockets in the system generally. The advantage of the "bypass" over the "batch" method will be apparent on reference to Figs. 1 and 2, showing the conditions of the oil in service.

Purifying Plant. Three types of plant are available for performing the duty required, namely, settling tanks, filters, and centrifugal purifiers or separators.

Settling Tanks. These occupy considerable space, are slow in action, and inefficient. Owing to the very slight difference in specific gravity between certain types of foreign matter and the entraining oil, it is nearly impossible to obtain a sharp separation between impurities and oil, and a certain proportion of the dirt must be withdrawn with the settled oil, or a considerable amount of oil will be wasted at each treatment. Where water is present in the form of an emulsion, it is almost impossible to remove it. Application of heat assists, but at the same time causes convection currents which tend to keep impurities in suspension. Settling tanks are seldom used alone for these reasons, but are occasionally found in conjunction with filters.

Filters. Certain limitations are inherent in filters. First, to remove fine impurities, the oil passages must be very small, therefore the filtration rate must be slow, or the filter must be bulky and costly. Second, as the adventitious matter must be collected on the feed side of the filtering material, the passages become progressively clogged, so that the rate of throughput is not constant. Third, it is practically impossible to remove water from the oil, especially if present as an emulsion, without special treatment. Many filters designed to

minimize these limitations are now available, but, as stated before, they are inherent in all forms of strainers and filters. On the other hand, certain types of filter can remove extremely fine particles of solid matter from oil, and where ultra-fine separation is desired they are very efficient. They are thus useful for batch purification, where the rate of treatment is not as important as in the continuous system.

Centrifugal Separators and Purifiers. These machines take advantage of the difference in specific gravity between the oil and the impurities to effect separation. Under the action of centrifugal force this difference is magnified several thousand times, and it is thus possible to obtain practically instantaneous and sharp separation between the clean oil and the adventitious matter. Conditions of treatment being equal, the factors mainly affecting efficiency in machines of this class are accentuation of the difference of specific gravity, deposition distance, and duration of treatment. Most centrifugal purifiers run at from about 10,000 to 6,000 r.p.m., depending on the size of bowl, and have a short distance of deposition. This is obtained by fitting the bowl with a pack of conical plates or disks, separated by six or eight ribs. The dirty oil is fed to the spaces so formed between the plates, which thus provide a large number of individual separating chambers. Water and solid matter travel outward to the wall of the bowl and are discharged into a trap, whilst the purified oil is forced towards the axis, and discharged into another trap for return to the main lubricating system.

Rate of Treatment. As mentioned earlier, the ideal plant would have a capacity equal to the rate of circulation, so that all bearings could be fed with oil direct from the purifier. As this is usually impossible, a compromise must be effected and the capacity of the purifier required is therefore calculated as a percentage of the quantity of oil in circulation, and not on the rate of circulation. The actual percentage depends on factors outside the control of the centrifugal manufacturers. The design of engine, the nature and quantity of the impurities formed, and the quality of the oil, are the most important. The engine designer should take care that no pockets are formed in the lubrication system where impurities can collect. If the separator is to do its work properly, the impurities must be fed to it. The usual practice is for the main lubricating pump to take its suction some distance from the bottom of the drain tank or sump, and for the purifier to be fed from the lowest point. A running balance will be struck between the rate of formation of impurities and the rate of removal in the centrifugal purifier, and therefore the greater the

capacity of the purifier the lower the impurity content of the oil fed to the bearings. When separators were first introduced, their true value was hardly realized. As a result, many plants were fitted with purifiers of too small a capacity, and even today the purifier is often an afterthought, instead of being considered as one of the most important items in the lubrication system. What is the value of a carefully designed system, expensive pumps, coolers, etc., and a high-grade oil, if, under working conditions, the oil is used as a medium for carrying abrasive matter, sludge, and water to the bearings? A margin must also be allowed for abnormal conditions. For instance, if a purifier with a capacity just sufficient to keep the oil in good condition in normal circumstances is fitted, it will not be large enough if excess water gets into the system through a fractured pipe or a leaky joint or gland. Then the engine must be stopped to effect repairs. On the other hand, faults of this nature have occurred, but the engines have been kept running for a considerable period, because the separators were large enough to keep the water down to a reasonable amount. Too much emphasis cannot be laid on the importance of fitting purifiers of ample capacity.

Installation. The normal installation takes its feed from the lowest portion of the drain tank or sump. It is strongly recommended that the oil should be fed to the separator by a pump driven from the separator motor. If the machine is installed above the sump level the pump is essential. Where the plant is below the sump level, gravity feed could be used, but it is dangerous. Should any fault occur, causing the separator to stop, then with gravity feed there is a danger that the sump will be drained before the fault is noticed and the feed shut down. The pump should be fitted on its discharge side with a spring-loaded non-return valve, the tension being such that the valve will close against the static head of oil, but opens under the combined head and pump pressure.

From the pump the oil is usually passed through a heater, either steam or electric, the temperature of treatment being from 120–160 deg. F. The lower figure is usually sufficient for turbine oils unless tight water emulsions are encountered. Heating the oil reduces the viscosity, thus lowering the resistance to the passage of solid particles through the oil. It also enables emulsions to be more easily broken down and so increases the permissible rate of feed for satisfactory purification.

There seems to be at the present time considerable concern about the type of electric heater used for lubricating oils. In some cases, indirect heaters are specified, i.e. those in which the element is im-

mersed in water, and the heat is transferred through tubes from the water to the oil. Whether this elaboration is justified or not is doubtful. Electric heaters of the low-loading type, 5–8 watts per sq. in. of surface, and 3,500 amp. per sq. in. of the cross-section of the wire, all steel, or steel and cast iron construction, have given no trouble in the author's experience except through careless handling. This can be guarded against by thermostatic control, which is practically essential with the indirect type. High-loading blade-type heaters with copper sheaths must on no account be used for direct oil heating on lubrication systems. They are liable to cause local overheating, with resultant partial breakdown of the oil and formation of carbon and acid sludge, especially where emulsions are present.

A small water feed is often introduced to the machine with the oil, especially if the oil is dry. The water tends to sludge out solids which would otherwise accumulate in the bowl, and so lengthens the period between cleaning. It also assists in reducing the amount of water-soluble acids in the oil. This water feed should be at the same temperature as the oil to be treated.

De-aerating devices are sometimes insisted upon for treatment of the purified oil, before return to the main circuit. The amount of air entrained in the oil as normally returned to the system is negligible compared to the aeration caused by the design of the lubrication system itself.

Treatment. There still exists some confusion of thought on the results which may be expected from centrifugal purification. It must be clearly realized that the centrifugal separator is a mechanical device for extracting from oil solids in suspension, and separating immiscible liquids, the fundamental condition being that there exists a difference in specific gravity between the oil and the suspended solids or immiscible liquids. During the last few years new methods of producing high-class oils have been introduced, such as solvent extraction, and the addition of inhibitors. These are undoubtedly a great advance on the older "straight" refined oils, but it has yet to be proved that it is possible for them to resist water, air, high temperature and pressure, indefinitely, when pumped round at high velocity. With the modern high-power 3,000 r.p.m. turbine, gland leakage is a problem to be seriously considered. Sooner or later, the unsaturated bodies, little as they may be in amount in these modern oils, will link up with the oxygen in the air and water, and form sludge. At low temperature the sludge is precipitated, but certain types become soluble at a higher temperature, say 120 deg. F. Oil is usually treated at 120–160 deg. F., in order to reduce its viscosity, thus making it easier to separate solid

matter and also to break down any water emulsions formed. At this temperature, it is obviously impossible for the purifier to remove this type of sludge, as it is in solution. It is usually precipitated in the coolers, and deposits in the oil ways and on the cooler tubes, thus causing the temperature of the oil to rise. There are various methods of overcoming this trouble. One is to fit two purifiers, one on the hot side of the cooler and one on the cold side. The first machine removes water and the usual solid matter, and the second the sludge. A second method is to fit an extra large machine on the cooled side, and run it at a slow rate of feed. Where the main engine is shut down at reasonable intervals, it is often possible to use the one purifier on hot oil when running, and on cold oil when stopped.

During the course of the breakdown of lubricating oil, apart from sludge, organic acids are slowly formed, and it has been the practice in the past to judge an oil, often entirely, on its rate of acid formation. Generally the rise in acidity and the rise in sludge content go hand in hand, and this has been quite a good method of keeping a check on the life of the oil. However, to-day, with some of the new solvent-extracted oils on the market, this does not always follow. This rise in acidity, although not actually harmful from a corrosion point of view, does shorten the life of the lubricant, as the acid effect is cumulative. The modern centrifugal separator is also able to play an important part in the removal of acids. The acids formed in this type of oil, are partially soluble in water and partially volatile. The volatile acids can be removed by efficient bearing and pipe-line vents, and the water-soluble acids can be removed at the point where centrifugal separation is carried out. When acid formation is met with it is always worth while to carry out tests in order to establish the proportion of water-soluble acids, and the rate and temperature of the water feed necessary to remove them.

It would be to the advantage of all parties, particularly that of the operating engineer, if closer co-operation existed between engine designers, oil suppliers, and manufacturers of centrifugal separators. The centrifugal separator can only perform its duties satisfactorily if it is supplied with the oil to be treated under conditions suitable for the removal of the impurities.

THE LUBRICATION OF LOCOMOTIVE CYLINDERS ON THE FRENCH RAILWAYS.

By M. Chatel*

The methods of lubrication employed for locomotive bearings, depending on the theory of the oil film, cannot be applied to the lubrication of the cylinders. There are many reasons for this, such as: the surface requiring lubrication is large; the relative speeds of the two contact surfaces vary in extent and sign at a frequency of as much as 800 oscillations per minute; the steam in contact with the surfaces may be at a temperature of 400 deg. C.; the nature of the work of the organs prevents them from being made from those metals which give the best coefficient of friction; while the special conditions under which locomotives work, such as running with the regulator closed, impose special requirements. In consequence, boundary lubrication, despite its known defects, has to be resorted to.

Piston Rings and Cylinders. The French railways have not yet come to a final decision about the nature of cast irons for piston rings and cylinders, though their mechanical qualities and the methods of testing have been laid down. The cast iron must have a homogeneous pearlitic structure, be free from ferrite, and have a scattering of cementite, but no phosphorus eutectic or manganese sulphide. The percentage composition, though left to the foundry and not specified, is, on the average, roughly as follows: total carbon, 3–3·3 (including graphitic carbon, 2·45; combined carbon, 0·75); silicon, 1·5–1·7; manganese, 0·5–0·7; phosphorus, 0·25; sulphur, 0·10.

The use of special alloying elements (silicon, nickel, and titanium or chromium-vanadium) is not obligatory, though often recommended.

The following tests are made: (1) Slow-bending test on a specimen $10 \times 8 \times 40$ mm., placed flat on two supports 30 mm. apart; the breaking stress must exceed 550 kg. for the "F.S.1" type and 650 kg. for the "F.S.2" type. (2) The shearing test must show a strength over 25 kg. per sq. mm. for the F.S.1 and 30 kg. per sq. mm. for the F.S.2 cast irons. (3) The Brinell hardness number must be between 170 and 240 for the F.S.1 and between 200 and 270 for the F.S.2 cast irons. These tests can be rounded off by measuring the modulus of elasticity on the Le Belland and Serin machine; a figure of over 12,500 should be obtained. As regards the comparative hardness of cast irons for piston rings and for cylinders, no definite decision has been arrived

* French State Railways.

at, but the tendency is towards the use of a harder metal for the rings.

To obviate the difficulty inherent in the systematic measurement of wear in service, it is proposed to install a special test bench which will simulate the work of the piston rings and cylinders under definite conditions of temperature and lubrication.

Cylinder Bore. The diameter of a cylinder, measured at any two points, must not vary by more than 0·3 mm. and the surface must not show tool marks. This result is obtained by the use of machines in good condition, provided with broad-nosed tools or tungsten carbide tools.

Piston Rings. There is a growing tendency to use numerous, springy piston rings. Modern machines usually have four rings per piston, each ring when new being usually 20 mm. thick, though the thickness in certain locomotives has been reduced to 15 mm. as an experiment. The rings are of the Carels type, with or without a hole for balancing, and are cut straight or obliquely. Each ring is adjusted in the piston groove to have, at normal temperature, a maximum clearance between the edges of 0·1 mm. against 1 mm. in the groove. The edges of the rings are rounded off so as to prevent wear of the cylinder by sharp corners and the surface is sometimes polished with coarse emery cloth; milling is not yet employed.

Special instructions are given on each railway regarding methods to be followed when replacing pistons and rings in rebored cylinders.

Cylinder Oils. Standard specifications have been agreed to for oils for machines without superheat or with moderate superheat. Conditions are more critical for powerful locomotives with high superheat, in which the pressure and the temperature at entry into the cylinder are equal to or exceed 16–17 kg. per sq. cm. and 370 deg. C. respectively. As fluid lubrication is impossible under these conditions, attention is paid particularly to the wetting capacity and the oiliness of the lubricant. It is essential that the film of oil be formed rapidly and uniformly, and that it should adhere well to the cylinder wall. The tendency to wetting, together with the value of the interfacial tension between oil and cast iron, depend on the chemical composition of the lubricant. Pure mineral oils spread well on metal without there being apparently anything more than a mechanical intermolecular adhesion. Animal and vegetable oils, like compounded oils, adhere more strongly, as the molecules attached to the surface form a kind of protective layer, though spreading is weak and soon stops.

Under the conditions in a locomotive, the viscosities of the oils show little difference at cylinder temperatures, so that the viscosity of an oil is not of prime importance, while "oiliness" is a quality of too indefinite a character for it to be introduced usefully into a specification. Under these conditions all that is done is to select an oil which can easily be atomized in the cylinder, i.e. an oil of average viscosity which has sufficient oiliness so that the film formed on the cylinder wall will be sufficiently resistant to high local pressures. A suitable viscosity must not be obtained by adding asphalt, as this would cause the rings to stick.

The reason why the oil withstands temperatures above its fire point is that the atmosphere in the cylinder is normally not of an oxidizing nature. The high temperature, however, causes some slight evaporation of the oil, whilst entries of air into the cylinder cause partial carbonization which can prove harmful in time. This risk can be decreased if an oil with a high fire-point is utilized and if other precautions, which will be dealt with later, are taken.

Guided by these considerations, the French railways have chosen for locomotives with high superheat a number of oils which are fairly similar one to another. A few analyses are given in the following table:—

Oil	A	B	C
Specific gravity at 15 deg. C.	0·904	0·900	0·905
Closed flash-point, deg. C.	278	295	270
Fire point, deg. C.	330	350	325
Viscosity at 100 deg. C. in centistokes	44	39	61
Viscosity at 200 deg. C. in centistokes	5·2	4·9	$15_{6\cdot 8}$
Setting point, deg. C.	+9	+3	—
Acid number	0·3	0·3	1
Saponification index	3	5	
Impurities	Nil	Nil	Nil
Asphalt, per cent	Nil	Nil	<0·05
Ash, per cent	Traces	Traces	0·015

These three types are practically all the same, but oil A costs more than oils B and C, so that the cheaper oils are being used in increasing quantities.

Methods of Lubrication. The oil must be distributed in such a way that, while its qualities are preserved, it arrives continuously and as uniformly as possible at the surfaces which require lubrication. This was first accomplished by the condensing lubricator, which is simple to install; the oil reaches the cylinder in the form of fine drops mixed

with steam and is satisfactorily distributed throughout the cylinder. Unfortunately the supply is irregular, as the oil can only enter the cylinder if the steam pressure in the piping is greater than that in the cylinder. In old machines there was a distinct loss of pressure between the boiler and the cylinder, but, in modern machines, the loss is much smaller. Thus, lubrication is properly carried out only if the regulator is partly open, thus causing a further loss of pressure. The pressure in the oil pipes can, however, be increased by fitting the pipes with capillary nozzles (Bernard-Poncet) which control the delivery. Various other defects of this system of lubrication can be somewhat mitigated by attention to discipline. For this reason and because of its simplicity, certain French railways still use the system on secondary machines with moderate superheat.

Automatic pressure lubrication has now replaced the older system on modern locomotives working under high pressure and high superheat. The oil pumps are of various makes and are driven from the driving mechanism of the locomotive, so that the oil supply is regular and certain, under any conditions of steaming. Under this system the oil is supplied to all the points, including the axleboxes, which require lubrication. The following points require attention:—

Position of the Pump. The oil must be sufficiently warm when fed to the pump, and this can be ensured by mounting the pump on the firebox backplate, in the cab, though this leads to long complicated piping which easily chokes. A better method is to mount one or more pumps on the footplates as near as possible to the cylinder and to supply steam under control to preheat the oil. With this system the arrangement of the drive of the pump is easier.

Control of Delivery. When the lubricator is mounted in the cab, the driver has a set of sights and controls in front of him and can see if oil is being delivered properly to all the points to be lubricated. Actually this control is illusory, as the sights become covered with coal dust and too much attention is required in the observation of signals, etc., for there to be time to attend to the lubricator. If the lubricator is mounted on the footplate, the provision of sights and controls in the cab will be expensive and awkward. For this reason it is usual to provide oil tanks having partitions, each small tank formed in this way supplying a set of pipes. The state of affairs can then be seen at a glance and this is sufficient in practice.

Pipework. The layout of the pipework of a lubricating system is a fairly delicate matter. Given a number of starting points, the number of points to be lubricated can be increased by providing branches. Such a method is a makeshift, however, as it is difficult to balance the resistance offered to the oil by the branches, and therefore difficult to

ensure proper oil supplies. Therefore the only pipes which should be duplicated are those supplying secondary points. In practice, however, some concessions are made. For example, in machines such as the Nord "Pacifics", where there are soft whitemetal packings, the lubrication of these can safely be neglected. Cast iron packing, however, must be lubricated. The high-pressure cylinder can be fed only at one point on the upper surface, where the steam is at a lower temperature. At a pinch, one can count on the oil contained in the inlet steam from the high-pressure cylinder and only lubricate the distributors. The author recommends that the high-pressure cylinders should be lubricated towards the dead-centres, where the piston is almost stationary, so that the piston rings can receive plenty of oil.

Distribution of the Oil. Non-return valves should be fitted as close as possible to the lubrication points to prevent steam from flowing back to the pump and to prevent oil from being drawn from the pipework when the inlet pressure in the cylinders falls. Otherwise, when the regulator is reopened, lubrication would only begin when the pipework had filled with oil again, which would require a run of some kilometres. The non-return valves are often combined with devices to supply saturated steam which surrounds the drops of oil and prevents carbonization.

Running with Closed Regulator. In this case, despite the air valve and bypass arrangements, the cylinder can suck back hot, dusty gas from the smokebox and heat it during compression. Most of the French railways inject steam or water into the cylinder exhaust to reduce the temperature of the gas and an automatic device to apply injection when the driver closes the regulator is under consideration. The same result can be attained by forbidding the driver to close the regulator completely, but close control of the men would be required.

Tests of Mechanical Lubricators. The chief tests applied to oil pumps are designed to ascertain the resistance and porosity of the cylinders at 250 kg. per sq. cm., and to measure the volumetric yield at 12 deg. C. with oils of varying viscosity at full speed and half speed, the pumping pressure being held at 200 kg. per sq. cm. The minimum volumetric yield for the maximum speed of rotation should be 0·8 and 0·75, according to the oils employed. Similar measurements are made at 6 deg. C., at which temperature the yield should be about $\frac{8}{10}$ of that at 12 deg. The regulation of the flow is tested and a fatigue test lasting 1,500 hours is carried out at about 12 deg. C. on special oil for superheated machines, the pumping pressure being

200 kg. per sq. cm., and the maximum service speed being maintained. After the fatigue test, the volumetric yield should not be less than 0·95 of the volumetric yield of a new pump.

Oil Consumption. The oil consumption varies greatly according to the nature of the traffic and the line. The minimum consumption is 5 grammes per kilometre and the maximum 15 grammes. So far it has not been possible to find out whether abundant lubrication reduces the cost of upkeep of the cylinders and piston rings and whether the balance of costs favours one method or another. This remark also applies to the oil, though it appears that high-grade oils of suitable characteristics are comparable and that the cheapest amongst the high-grade oils will suffice. For equal power, simple expansion machines require a little less oil than compound engines, which have more points requiring lubrication.

Wear. For modern machines, high-duty cast iron is required, though the most suitable compositions to use are not known with certainty. The piston rings are replaced as follows:—

Passenger engines (Pacific types):—
High-pressure cylinders, every 20,000–40,000 kilometres.
Low-pressure ,, ,, 45,000–60,000 ,,
Goods engines (Consolidation, Decapod, Mikado types):—
High-pressure cylinders, every 30,000–50,000 kilometres.
Low-pressure ,, ,, 30,000–70,000 ,,
Simple-expansion cylinders
(Mikado) ,, 40,000–50,000 ,,

Comparison between the above figures is somewhat difficult as so much depends, particularly for goods engines, on traffic requirements. There have been isolated cases where the piston rings had to be replaced after 10,000 kilometres, but these were exceptional.

INSTALLATION AND OPERATION OF CENTRIFUGAL OIL SEPARATORS

By R. H. Dolton and H. Mackegg*

The centrifugal separator was first used in Great Britain for the purification of lubricating oil about twenty years ago. At first this system was used for the purification of turbine and Diesel engine lubricating oil in land power stations and on board ship, but later, applications for general industrial work were developed, and centrifugal separators of special design are now used to meet the individual requirements of specific lubrication problems in industry. The method of

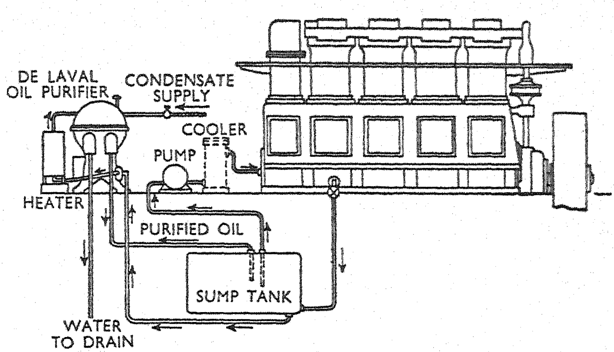

Fig. 1. Installation for the Continuous Bypass Purification of Diesel Engine Lubricating Oil

installation and operation of centrifugal separators for different purposes varies considerably, and the object of this paper is to summarize many years' experience in this field.

Diesel Engines. The correct method of centrifugal purification of Diesel engine lubricating oil is of paramount importance and has, during recent years, been closely investigated. It has now been established that by the provision of oil heaters of ample size and centrifugal purifiers of high purifying capacity it is possible to keep the circulating oil free from abrasive impurities, carbonaceous sludge, and water. The general arrangement of a typical Diesel lubricating oil purification installation is shown in Fig. 1, from which it will be seen

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that the lubricating oil is pumped from the lowest part of the circulating system and passed through the heater and thence to a centrifugal purifier of adequate capacity, and returned to the circulating system. The returned oil may be passed through a cooler if required. In this system, which is comparatively simple to include in an engine room layout, the oil must be passed to the centrifugal separator at a temperature of 160–180 deg. F. and the heater, therefore, should be adequately rated to perform this duty at the full throughput capacity of the separator. Further, the centrifugal purifier must have a high separating capacity which will enable the whole of the lubricating oil in the circulating system to be effectively purified at least once every four hours. Observance of these conditions make it possible to purify the circulating oil efficiently at a high rate of flow, so that the bulk of the oil in circulation is kept in a constantly good condition irrespective of the rate of, or reason for, the degree of contamination of the lubricating oil during its passage through the main lubricating system of the engine.

In Diesel engines fitted with oil-cooled pistons, the cooling oil can be continuously passed directly from the piston cooling system to the centrifugal purifier. In this particular system no preheating should be necessary, and it should be emphasized that continuous purification is essential.

For Diesel installations of small power, centrifugal separators having capacities proportionate to the amount of oil in circulation can be used on the continuous bypass system, with or without the addition of a fine filter. Where a filter is also used, the centrifugal separator will remove water and sludge, thus increasing the efficiency and life of the filter.

In installations where, owing to the design of the engine, it is not possible to install a centrifugal separator on the continuous bypass system, it is possible to install a batch treating plant. In such a case an overhead tank for dirty oil is required. This tank should be fitted with heating coils and a draw-off cock at the bottom of the tank for sludge removal. The temperature of the oil should be raised to 180 deg. F. and, after a short settling period, the oil can be effectively purified in the centrifugal separator, and collected in a clean oil storage tank for future use.

Running-in of Internal Combustion Engines. The clean lubrication of internal combustion engines during running-in on the test stands is obviously of extreme importance and greatly influences the condition, efficiency, and life of the engines. The interior of a newly assembled engine will inevitably contain small metallic particles, the result of machining, also possibly moulding sand, and unless precautions are

taken to remove these abrasives from the lubricant they will naturally have a detrimental effect on the finish of the bearings, pistons, and cylinder bores. To maintain the lubricating oil in good condition it will suffice to arrange for continuous centrifugal purification of the oil during running-in and test period. With automobile and aero-engines it is not generally possible to install a centrifugal purifier for each individual engine and the layout illustrated in Fig. 2 is therefore recommended. In this system, clean oil is continuously pumped through the

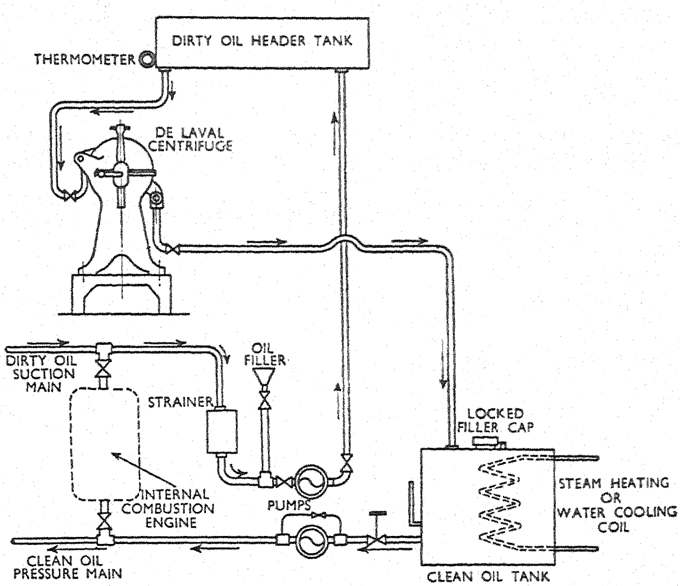

Fig. 2. Arrangement of Oil Purifying Plant for Internal Combustion Engines Running in Stands

engines and delivered to an overhead tank, from which the oil passes to the centrifugal separator, where solid impurities, water, etc., are removed, and thence to the clean oil tank. Heating and cooling coils are fitted to the clean oil tank so that the temperature of the oil passing to the engines can be maintained constant independently of the air temperature. The rate of flow of clean oil to the engine is, of course, controlled by the engine pump itself. This method ensures that purified oil is pumped at all times to the engines and is to be preferred to the system in which one common oil storage tank is used and the oil is purified on a separate bypass circuit.

Steam Turbines. The method of installation of an oil purifier for a steam turbine will largely depend on the type and size of turbine and the grade of lubricating oil which is used. In some cases the oil is not subjected to very heavy duty and the purifier is merely required to remove adventitious impurities such as iron rust, etc., in addition to traces of sludge which may be formed during service.

With the development of the modern large steam turbine, with higher bearing pressures, increased steam pressures, and hence higher bearing temperatures, the turbine designer has been confronted with the great problem of efficient lubrication and associated with this, the problem of maintaining the lubricating oil in good condition during use. The deterioration of turbine lubricating oil is an oxidation process caused by the extreme conditions of temperature and pressure to which the oil is subjected in the presence of water. The result is that organic acids are formed, which in turn attack the metallic surfaces with which they come into contact and form metallic soaps. These metallic soaps accelerate the formation of sludge and emulsions.

The authors' experience is that these acids tend to be soluble in water at the time of formation and can be removed with the water which is generally present in turbine oil (from steam leaks, etc.) or by independent washing of the oil with water. To maintain the oil in good condition, therefore, continuous separation of water and sludge is necessary, and this can be effected by means of a centrifugal separator connected to the turbine oil tank and operated on a bypass circuit. A layout of a typical installation is illustrated in Fig. 3. The oil is pumped from the turbine oil tank through an indirect water-jacketed heater, and thence directly to the centrifugal separator, at which point the oil should have a temperature of approximately 140 deg. F. If the oil temperature is lower than this some difficulty is experienced in obtaining efficient separation, whilst if the temperature is considerably higher there is a risk that the sludge may go into solution. At the inlet of the purifier a stream of hot water (1–2 per cent of the volume of the oil) is fed in with the oil. This water can conveniently be taken from the jacket of the heater. Thorough mixing of the oil and water occurs at the inlet of the centrifuge bowl itself. This washing effect ensures the removal of all water-soluble acids and, provided the purifier is of adequate size and is operated continuously, the accumulation of acid in the oil is reduced to a minimum.

In certain instances there is no leakage of steam into the lubricating system, so that the lubricating oil is quite dry. If in these circumstances there is also no sign of appreciable development of acidity, there is no need to wash the oil with water, but nevertheless the installation should be so arranged that this process can be employed if required.

It is difficult to make any general recommendation regarding the capacity of the oil-purifying unit for any particular size of turbine owing to the varying conditions of operation and design. For any given turbine rating some turbine builders use a larger oil reservoir than others, but the authors believe that it is best to rate the size of the purifier on the quantity of the oil in the system. Thus the purifier should be capable of passing the entire contents of the turbine reservoir in 4–8 hours, and the centrifugal separator should be kept in continuous operation at all times when the turbine is running. This ensures that the various impurities present in the oil are removed immediately after they are formed.

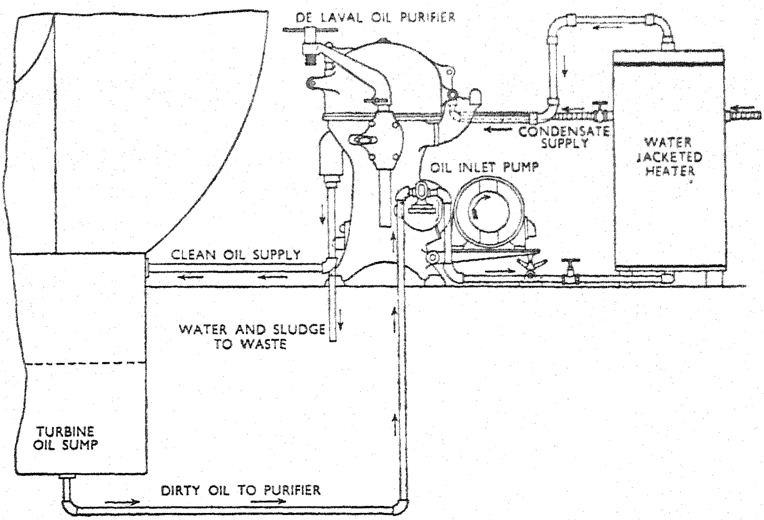

Fig. 3. Centrifugal Oil Purifier coupled to Turbine for Continuous Bypass Purification

Machine Shop Oils. It is the general practice in machine shops to recover waste cutting oil either by draining or by employing a basket type of centrifuge. The oil so obtained contains in suspension finely divided metal particles and carbon particles formed through overheating, water, and bacteria. These impurities impair the oil, damage the machine tools, reduce the quality of the work turned out, and may in extreme cases cause dermatitis among machine operators.

The installation of a centrifugal separator enables this oil to be purified and renders it suitable for use again. The dirty oil is generally collected in an underground storage tank from which it is pumped through a heater which raises the temperature to 180 deg. F. and not

only destroys all harmful bacteria but also reduces the viscosity of the oil. The oil is then passed to the centrifugal separator where all solid impurities and water are continuously removed, giving a purified oil which can be used alone or mixed with new oil.

Another important application for centrifugal separators in machine shop practice is for the treatment of the paraffin oil used as a lubricant in cylinder honing machines. These machines are fitted with a sump to hold the lubricant which is continuously pumped on to the honing tool during operation. The oil becomes contaminated with fine grinding dust, metallic particles, etc., which must, of course, be removed if high-quality work is to be produced. Complete purification of this oil can be obtained by circulating the oil through a centrifugal separator on a bypass circuit. The same system is also used for the purification of cutting oil for large automatic machine tools.

Conclusion. The foregoing notes indicate a few of the uses of the centrifugal separator in the maintenance of lubricating oil at a high degree of purity under exacting working conditions. It is obvious that however carefully the machine builder may design his lubrication system it cannot function satisfactorily unless the lubricant itself is free from impurities and is constantly held in that state. As the centrifugal separator can continuously separate and remove both solid impurities and water from lubricating oils of all kinds the machine designer is thus provided with a means of increasing the efficiency of any lubrication system.

RATING OILS FOR CYLINDER WEAR BY THE IRON CONTAMINATION METHOD

By Professor H. A. Everett * and G. H. Keller †

Early in 1933 the authors undertook extensive tests of a series of oils to investigate their performance in engines under operating conditions. The engines were of the automotive type, six cylinder, $3\frac{1}{8} \times 4\frac{3}{8}$-inch, 75 rated b.h.p., running at 3,600 r.p.m., full-pressure lubricated, (30 lb. per sq. in.); and had aluminium pistons. To minimize individual mechanical idiosyncrasies, four identical units were used for each test run. These engines were all new and were put through the same "life cycle" as regards running-in and subsequent operation. Runs on a test oil were preceded and followed by runs using a reference oil.

One important part of the programme was the investigation of the wear permitted by the different oils. As wear determinations by the mechanical measurement method involved periods of operation far in excess of what could be considered, a method was developed independently of other experimenters for determining the amount of wear occurring in the cylinders and rings by progressive analyses of the iron contamination of the crankcase oil. No claim is made to priority, as others were working along similar lines (Boerlage and Gravestyn 1932), but the authors' development was independent of, and without contemporaneous knowledge of, the work of the others.

The data obtained by this method appear to be remarkably reliable and sensitive. In fact, when plotting data obtained in earlier work there were occasional large variations which were disturbing as indicating a possible unreliability. These apparent discordances later were traced to slight operating or mechanical changes, the effects of which, when plotted on the exaggerated scale inherent to the sensitivity of this method, were strikingly apparent.

In the method, samples of crankcase oil are drawn at periodic intervals from the oil supply line between the pump and the bearings, thus ensuring a sample representative of the oil actually being used for lubrication. The analyses of these samples, plotted on time elapsed, give a curve of progressive iron contamination, from which can be found the instantaneous rate of wear occurring throughout the run. Such a procedure is based on the assumption that most of the iron wear

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occurs between the cylinders and the piston rings and that such iron will be washed off by the lubricating oil, causing a progressive increase of iron contamination throughout the repeated circuits made by the lubricating oil. The method also assumes that oil which leaves the engine by combustion or otherwise, carries its iron with it. As to the first assumption, the progressive increase in the iron content of the crankcase oil is evidence that iron wear is carried with the oil from the cylinders to the crankcase. The second assumption was verified by experiments which showed that a minute but appreciable amount of iron went out continually with the exhaust gases. Iron has also been found in the carbon deposited in the combustion chamber. A complete account of the method of iron analysis and the results obtained in the first series of tests has been published (Everett and Stewart 1935).

The chemical analyses of the progressive samples were reported on a percentage basis, i.e. the percentage of iron found in each sample.

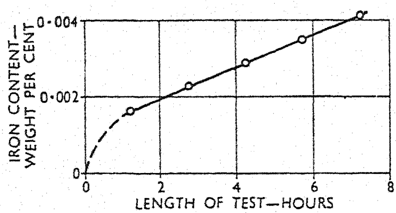


Fig. 1. Progressive Iron Contamination

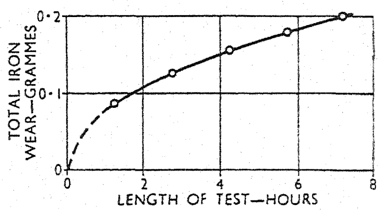


Fig. 2. Total Iron Wear

Provided no oil had been added or taken from the system throughout the test, it would have been necessary only to multiply these percentages by the total quantity of oil to obtain corresponding values of the total wear. However, the quantity of oil was continually changing due to consumption, samples, leakage and renewal, all of which had to be taken into account before a curve of total wear could be drawn.

Fig. 1 shows a typical curve of the progressive iron contamination obtained by plotting the results of the analyses returned from the chemical laboratory, against the time corresponding to the taking of the sample. From this curve the total iron wear could be obtained by multiplying the weight of the oil in the sump by its concentration, with an allowance made for the amount of iron removed from the system by the oil samples, by leakage, and by oil burned in the combustion chambers. The magnitude of each sample was accurately known and it was thus possible to obtain a curve like Fig. 2, which shows the total iron wear, plotted against hours of operation. The slope of this curve of total wear measured the instantaneous rate of wear.

The early tests lasted eight hours and no make-up oil was added during the run. Some tests were made of twenty-four hours duration in three periods of eight hours each and in these it was imperative to add make-up oil. Accordingly additions were made at hourly intervals,

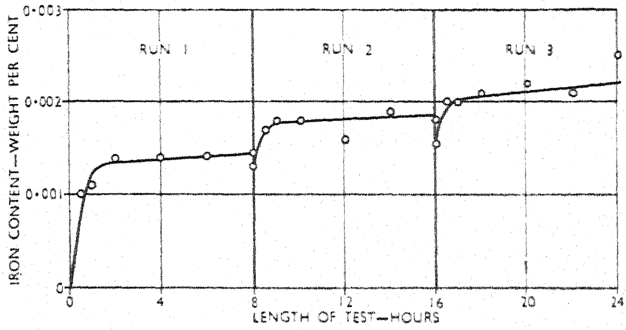


Fig. 3. Typical Result of Tests with One Engine

thus more nearly maintaining a constant quantity of oil in the sump. Fig. 3 shows a typical test for one engine. In these curves the laboratory analyses are plotted as determined without allowance for consumption, samples, etc. In Fig. 4 the curves of Fig. 3 are corrected to

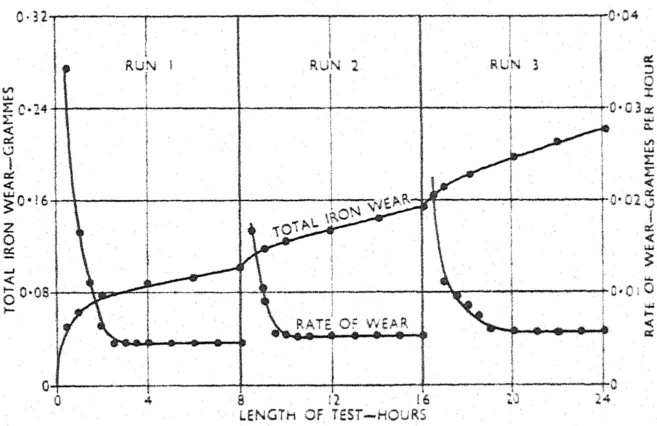


Fig. 4. Results Corrected to give Total Wear

give total wear as just explained, and the curves of rate of wear derived from these.

Originally, samples were taken at intervals of approximately one hour (Fig. 1). The bridging of the gap between the first point and the

origin was, however, open to conjecture. To settle the matter, samples were taken at approximately ten-minute intervals for the first two hours. The analyses showed iron contents progressively greater for each of these ten-minute intervals, and proved conclusively that the curve went through the origin, a conclusion in harmony with the theory that there should exist a high rate of wear on starting a cold engine. These engines were operating at speeds of revolution corresponding to a road speed of 60 m.p.h.; hence a time interval of 10 minutes corresponds to 10 miles of travel, yet this interval sufficed to permit an evaluation of the wear which had taken place, a most illuminating indication of the extreme sensitivity of this method.

A sensitivity as great as this is not an unmixed blessing, as engines are continually undergoing mechanical adjustments. Such apparently innocuous changes as the removal and replacement of a piston, show up as large irregularities in the wear curves. When a major change occurs such as the installation of new piston rings, the analyses show so large a contamination that the scale of the curves must be reduced. To illustrate, from one run during which no evidence of serious engine disturbance had been noted, the returns for iron contamination were about fifteen times their normal values. Check analyses confirmed the original data, and removal of the head showed scoring had taken place in one cylinder of the unit for which the high analyses had been returned. Fig. 5 shows the effect of the normal reconditioning which was customary between runs. This reconditioning consisted of the removal of oil pans, cylinder heads, piston, and connecting rod assemblies for cleaning, and their reassembly in as nearly as possible the original mechanical adjustment.

The data on starting wear are interesting. Returns are available from approximately 70 tests of from 8 to 24 hours operation, each using four engines operated under identical conditions. The curves for rate of wear were similar for all the tests and Fig. 4 may be considered typical of those obtained for the 24-hour runs. In these, the engines operated in three 8-hour runs, remaining idle overnight. From all these runs the data indicate that the initial rate of wear is very great but decreases rapidly during the first two hours, after which it assumes a value which may be considered normal for that operative state. This rate is not always exactly the same from day to day, possibly as the result of variation in thermal distortion or some other secondary influence. However, it should again be noted that large apparent differences are actually very minute quantities.

It is difficult to obtain an accurate correlation between determinations of wear by this analytical method and those obtained by actual ring or cylinder measurements. However, if the assumption is made that all

the iron found in the oil came from the ring and cylinder and from each equally, the foregoing analytical tests indicate a cylinder bore wear of 1/1,000 inch for each 15,000–20,000 miles of road travel. This wear is low compared with ordinary road results, but the engines were operated under conditions particularly favourable to minimum wear.

Tests have been made on fifteen different oils, all matched to the same viscosity, 217 S.U.S. at 130 deg F. Some of these oils were from

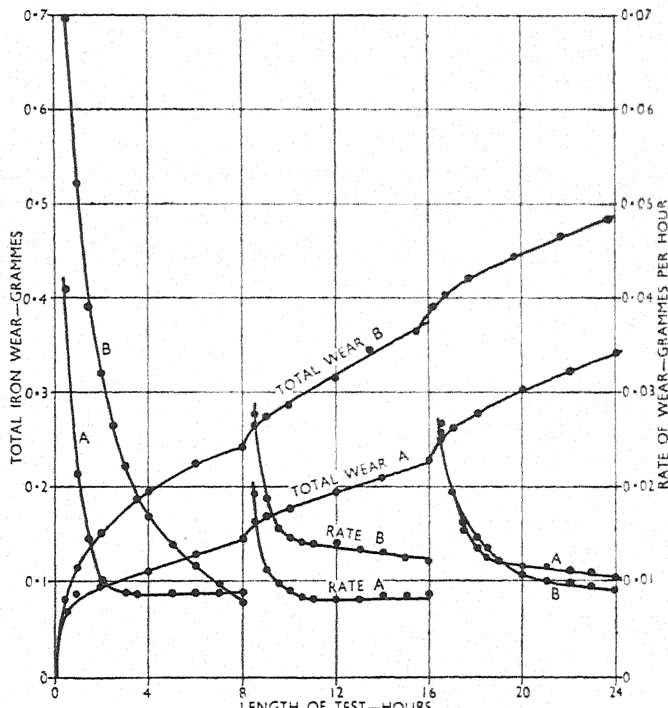


Fig. 5. The Effect of Normal Reconditioning on Wear
A Before overhaul. B After overhaul.

different fields or crudes, but most of them were from the Pennsylvania fields. Oils from the same field differed in the method of refining and also in the type of blending stock used. The accuracy of the analytical work gave wear determinations probably reproducible to about 3 or 4 per cent. Engine operating conditions rarely could be considered reproducible closer than 5 per cent, hence data for relative wear experienced when using different oils are not considered conclusive unless the differences in wear exceed about 10 per cent.

On this basis it was found that the wear when using an oil from a Western U.S. crude was materially higher than when using the reference oil, a conventionally refined oil from Pennsylvania crude. It was also found that oils from a Pennsylvania crude gave slightly different wear according to the method of refining, i.e. the solvent extracted oils showed a wear slightly higher than the reference oil during the first eight hours of test. The differences in wear, using oils which were different blends of conventionally refined Pennsylvania stocks, were insignificant.

In conclusion, it is believed that this method of determining cylinder wear gives results with reasonable periods of operation and can give instantaneous values of rate of wear. The mechanical methods of determining wear satisfy neither of these important criteria. The outstanding difficulty, however, is the supersensitivity of the method. The simple removal and immediate replacement of a piston assembly may cause the wear to become twice normal; or the installation of a new piston ring may cause the wear to become five times normal. Therefore notable changes in wear may be observed which are due to apparently trivial causes. The use of a method of such great sensitivity, therefore, must be guided by discretion and a complete knowledge of contemporaneous mechanical changes.

References

BOERLAGE, G. D., and GRAVESTYN, B. J. J. 1932–3 *The Motorship* (British), vol. 13, p. 171, "Cylinder Wear in Diesel Engines".

EVERETT, H. A., and STEWART, F. C. 1935 Pennsylvania State College, Bulletin No. 44, "Performance Tests of Lubricating Oil in Automobile Engines".

LANGDON, H. H. 1927 State College of Washington, Engineering Experiment Station, Engineering Bulletin No. 42, "Laboratory Methods of Comparing Lubricating Values of Automotive Oils".

SERVICE EFFECTS ON LUBRICATING OIL

By A. E. Flowers*

Two previous papers† give a description of the contaminants and changes in characteristics that had been observed in the lubricating oil in Diesel engines. Part I of this paper presents the results of several years' further observations on these same Diesel engines, while Part II presents the results of several years' observations on the contaminants and changes in characteristics of the lubricating oil in two geared steam turbines. All tests were made in accordance with the latest methods or precautions of the American Society for Testing Materials current at the date of making the test.

I. Diesel Engine Lubricating Oil

The Diesel engines, rated at 300 h.p. at a speed of 600 r.p.m., had six 10-inch diameter cylinders and 12-inch stroke trunk pistons. Solid fuel injection at a pressure of 3,000 lb. per sq. in. was employed and all parts were supplied with lubricating oil from a 54 (U.S.)-gallon crankcase by forced feed at a rate of 12 gal. per min. at an ingoing temperature of about 155 deg. F. This oil was purified continuously by a small centrifugal separator on a bypass at the rate of about 17 gal. per hr., equivalent to cleaning the system oil about every 3 hours.

Changes in the Oil. Fig. 1 shows the direction and amount of the changes in the usual characteristics of the lubricating oil. The values for new oil are represented by the initial points of each curve on the vertical axis. These curves also show the generally stabilized condition or balance between centrifugal purification plus make-up on the one hand, and service deterioration on the other. Broadly speaking the changes in the oil may be classified as "native" and "foreign", both contributing to produce solid sediments or sludges, which may then be removed by a centrifugal separator.

The "native" changes may be considered as those brought about by chemical reactions and polymerizing effects; the "foreign" as those due to extraneous materials and contaminants such as water, dilution from unburnt fuel, blow-by and exhaust gases, smoke, sulphur dioxide, etc., road dust from "breather" and combustion air, metallic particles left

* The De Laval Separator Company, New York.

† Trans. A.S.M.E., 1930, vol. 52, OGP-52-2, "Service Characteristics of Diesel Engine Lubricating Oil"; Trans. A.S.M.E., 1930, vol. 52, OGP-52-12, "Purification of Diesel Engine Lubricating Oil."

from manufacture or overhauling, and worn metal from pistons, rings, liners, journals, bearings, pumps, etc. These "foreign" materials as well as the end products of the chemical reactions exert in their turn a tremendous catalytic effect to accelerate further the chemical reactions. Iron and copper, particularly in the fine state of subdivision resulting from wear, each exert a specific catalytic effect on particular oils. If road dust and the colloidal carbon from cracking and blow-by exert a catalytic effect, it is largely masked by the overpowering catalysis of wear particles and chemical end products.

The oil itself, when completely separated from the sediments and end products may show only moderate changes in properties. Dilution, either from fuel or resulting from the light "cracked" end

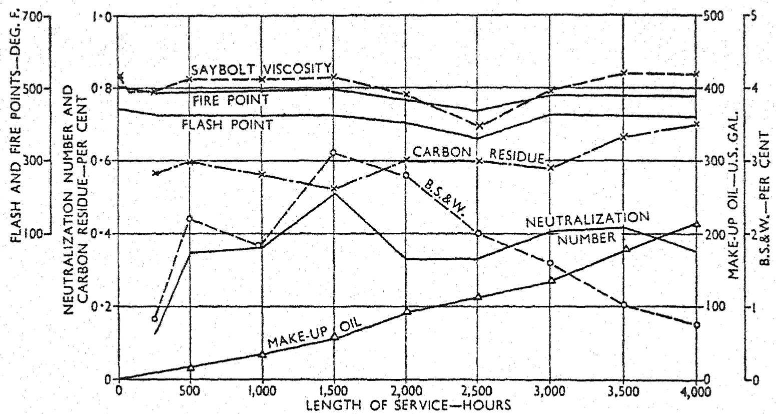


Fig. 1. Service Effects on Diesel Engine Lubricating Oil

The "B.S. & W." curve refers to bottom sediment and water.

products, tends to reduce the flash point appreciably and also the fire point, though to a lesser extent; and to reduce viscosity. Removal of dilution by evaporation or distillation may raise all three of these values, partly because the heavy end products from cracking are still present and because the oxidation products are generally more viscous than the original oil. There seems to be considerably more cracking than oxidation of Diesel lubricating oil, resulting not only in light and heavy end products, with the contrasting effects just mentioned, but also in the production of considerable colloidal carbon. Some colloidal carbon comes from the fuel as a result of blow-by.

Oxidation produces primarily a slow increase in organic acidity, best expressed by the number of milligrammes of potassium hydroxide per gramme of oil required for neutralization (the neutralization

number), which eventually reaches a fluctuating but relatively constant maximum, determined by the rate of purification and make-up on the one hand, and chemical reaction on the other. No definite corrosion effect has ever been proved to result from such values for the neutralization number, but the presence and magnitude of this type of acidity are generally considered as correlating with the oxidation that has taken place. On the contrary, this acidity may be one factor in the increased film strength of used and of well reclaimed oils, resulting in lower starting friction and wear. Corrosion, when it does occur, appears to be fully accounted for by the presence of water or sulphur compounds. In general, tests have shown that the remaining oil is as good a, if not a better, lubricant than when new.

Amounts and Character of Sediments and Sludges. These 300 h.p. engines produced 150–200 grammes dry (i.e. "oil-free") sludge per 100 hours of operation. The sludge as deposited held 45–60 per cent by weight of oil and weighed about 1·11 grammes per cu. cm. The "oil-free" sludge weighed 1·2 to 1·5 grammes per cu. cm. Continuous removal at about this rate brought the oil to a state of equilibrium, with regard to any further changes, in a few hundred hours, which state continued for the several thousand hours further running, without dumping or changing the oil, except for the make-up of about 2 (U.S.) gal. per 100 hours of engine operation, and a fluctuating but relatively constant amount and character of the sediments and sludges. The examination of these sludges reveals, however, vastly more interesting information than is given by the study of the oil itself. On a new engine, or after repairs, the sediment is largely (up to 75 per cent) metal or metallic oxides, some left by the machining operations and the remainder produced by the wearing down of high spots and rough surfaces. Later the sediment becomes stabilized at values that may be only 5–10 per cent. The principal constituent is colloidal carbon, which finally becomes constant at about 70 per cent. The remainder, 20–25 per cent, consists of gummy residues from oxidation and possibly cracking, and 0·5–2 per cent of air-borne road dust. The metallic particles and oxides, and road dust are certainly responsible for much continued wear. Their removal reduced the bearing wear on these engines from 0·001 inch in 10 weeks to a quantity too small to measure. The gummy residues may clog oil passages, particularly in places where bends or pockets occur or where local cooling exerts a continuous precipitating effect.

Sulphur from the fuel oil was traced in the oil and finally in the sludge. Sulphur, as such, is soluble in oil and would not be removable by centrifugal purification, but the sulphur dioxide produced by

combustion seems to form a compound with the other reaction products, resulting in a sludge which can be removed by centrifugal purification.

Carbon, in the highly dispersed colloidal state which it possesses at first, seems to have all the good properties claimed for colloidal graphite. It prevents wear, fills irregular surfaces, and lowers starting friction. The individual particles tend, however, to agglomerate with time, forming clusters or large masses that will settle or can be centrifugally separated. Such agglomerations should be removed. Water and heat accelerate this settling and it has recently been discovered* that heating under pressures of 5 to 10 atmos. with water will precipitate this colloidal carbon and other sludges completely, in a few minutes.

It should be emphasized that only agglomerated carbon needs to be removed, to avoid clogging lubricating passages and sumps and that complete removal of carbon reduces the so-called "film strength". In any case fresh carbon is continually being produced so that new oil acquires its quota quickly, and the catalytic effect it produces on oxidation reactions is therefore unavoidable.

Metals or metallic oxides should be removed as completely as possible. They cause serious wear and even scoring if left over from machining and fitting. As has already been pointed out, iron and copper are powerful catalytic agents in hastening oxidation and each reacts more powerfully on some oils than on others. If the content of metal increases in amount after an engine has once been run-in, this is an almost certain warning that undue wear is going on and some part, probably a bearing, is disintegrating and about to fail.

Road dust, stone, sand, etc., should also be removed as completely as possible and it is just such materials as "metals" and "sand" that are most effectively removed by centrifugal purifiers. However, it must be observed that fully satisfactory results cannot be had if the purifier is too small for the service conditions or if the oil is pumped through the purifier at so high a rate as not to permit the most thorough purification of the oil put through. Low-grade, poorly refined oils will also change so rapidly in service that no purifier could keep them in condition. Cases have been observed where a low-grade lubricating oil (suitable enough for ordinary bearings) was reduced to 90 per cent sediment, by volume, or 10 per cent free-flowing oil within three or four days and lay in the bottom of the crankcase in a livery mass.

II. Geared Turbine Lubricating Oil

The two steam turbines were built to operate at 3,600 r.p.m. on

* Patents applied for.

a steam pressure of 200 lb. per sq. in., each driving through a 6/1 herring-bone reduction gear a 600 kW. electric generator, running at 600 r.p.m. The turbine lubricating oil was fed by gravity from a common overhead tank holding 100 gallons with pump return and was purified on a continuous bypass system at 50 gal. per hr. One or the other of the two turbines was operated 9¾ hours daily for five days of the week and 4¾ hours on the half working-day Saturdays for the first three years, but on a reduced schedule for the remainder of the time. The new oil had a Saybolt universal viscosity at 100 deg. F. of 290 sec.,

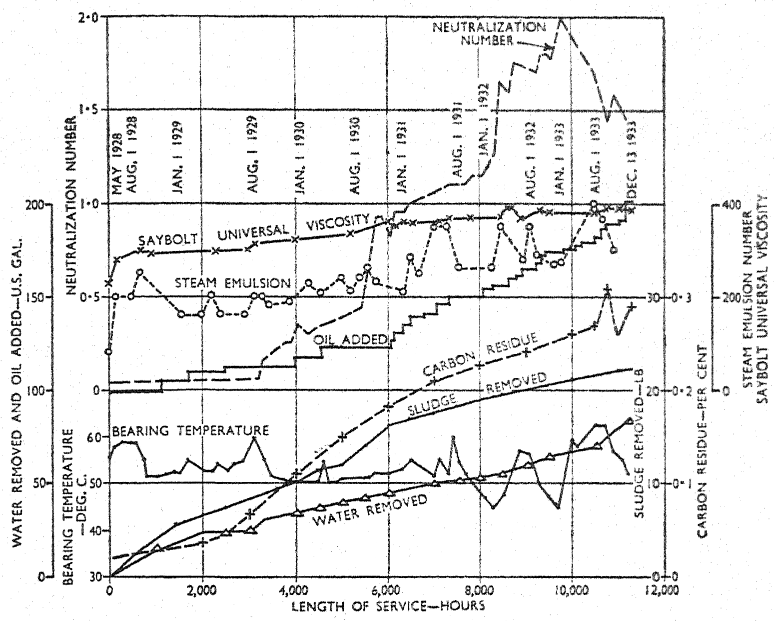


Fig. 2. Service Effects on Geared Turbine Lubricating Oil

a steam emulsion number of 153 sec., and a neutralization number of 0·01 mg. of potassium hydroxide per gramme of oil and was considered a well-refined, commercial grade oil. Fig. 2 shows the change in the principal characteristics of this oil for a period of over 11,000 hours, over 5½ years of operation, without change of oil, except for make-up, of which the total amount in the 5½ years was almost exactly equal to the original amount put into the system. The rate of make-up was least during the first years, increasing in the later period of the test, about one-quarter being used up in the first 5,000 hours and three-quarters during the second half.

Acidity. There was usually (Fig. 2) an increase in neutralization number (acidity) during the latter part of each summer, following to this extent seasonal temperatures. During the last 2,500 hours, the neutralization number reached a maximum of 2 mg. of potassium hydroxide per gramme, receded later to 1·35 and at the close was 1·75 mg.

Emulsification. The tendency to emulsify with water, as measured by the steam emulsion number, fluctuated between about 200 and 400 sec.

Viscosity. The viscosity of the oil increased not at all for the first year, but went up by about 17 per cent during the second year. Thereafter the increase was only 17 per cent more for the next 3½ years because during the last year there was no further definite increase.

Carbon Residue. The carbon residue increased steadily throughout the run, indicating that considerable oxidation effects were in progress.

Stability of Oil. It is possible that this oil had reached an approximately final condition at the end of the period of observation of 11,000 hours, at which time the make-up just equalled the original quantity, but while this might seem justified by the fairly constant values of the characteristics of acidity, emulsibility, and viscosity, the real trend may be masked by the slow rate of change. The oil consumption figures appear more significant if taken over sufficiently great periods of time, such as 2,500 hours, or about a year. This shows a usage of 15 gallons in the first 2,500 hours, only 10 gallons in the second, 28 in the third, 25 in the fourth, and 26 in the next 1,250 hours. This would seem to indicate some early high rate of loss, possibly due to evaporation, a lower rate next, and finally a doubled loss later, which was quadrupled in the last 1,250 hours. For these and other reasons it would seem that a stable state had not yet quite been reached.

Character of the Reactions. The principal reactions in turbine lubricating oil appear to be oxidation, in contrast to Diesel lubricating oil, where cracking seems to be the dominant factor. Oxidation is powerfully influenced by temperature, by the presence of water, and by natural or artificially added inhibitors. The role of temperature needs little further elaboration, but the effects of water appear less generally understood. Water comes from three sources. Some comes from condensation from the atmosphere on cold metal parts during shut-

down, some from direct solution* in the oil, the amount depending on the humidity of the atmosphere and the oil temperature, and a major part from steam leaks. There is no possibility of absolutely preventing water from getting into the oil, and unquestionably water aids in the oxidation reaction.

It is possible, however, to remove continuously, by means of a centrifugal separator, all *free* water. The removal of free water not only removes a dangerous element so far as lubrication is concerned, but, what is still more important, washes not only the sludge out of the oil, thus preventing its settling on the cooling coils and causing overheating, but also acts preferentially to dissolve the *freshly* formed acidic bodies which, when in the nascent state, are about ten times as soluble in water as in oil. Washing the oil† with 20 per cent clean, hot water, which is then completely removed by a centrifugal separator, will keep the acidity permanently to low values.

Water washing is inherent in all steam turbines to a greater or less extent, depending on steam leaks, condensation, etc. If water is absent for a long time, but suddenly gets into the oil through accident such as a leak from the cooling coils, very serious sludging often immediately occurs. The sludge and the water emulsion impede lubrication, cause heating of the bearings, and interfere with the heat transfer of the cooling coils, unless immediately removed.

The effect of air is important, but it is probable that there is no practical way to exclude it, and in any case the oil comes into service with some air in solution in it, quite often one-fifth of its own volume. As the oxygen in the dissolved air is exhausted by chemical reaction with the oil, more is quickly taken up from the atmosphere.

A secondary effect, resulting from oxidation and moisture, is the production of metallic soaps of iron, copper, tin, and lead. The source of the metal is, of course, the wear of journals and bearings. The worn metal particles are naturally in an extremely fine state of subdivision with, therefore, a large ratio of surface to volume or weight, and are, consequently, in the most favourable condition for chemical reaction. These metallic soaps are generally quite soluble in the oil and so cannot be settled or separated centrifugally. When oil finally gets into this state it should be removed from the system and put through a complete chemical reclamation process.

* Water is soluble in oil. A very highly refined oil will dissolve 10 to 40 parts per million of water at room temperature and many times this when hot. A poorly refined oil, and particularly those that are emulsifiable with water, may dissolve 500 parts per million at room temperature. It is this dissolved water which precipitates on cooling that is principally responsible for the fog that appears in clear *hot* oil upon *cooling*.

† Water washing, in accordance with the De Laval-Funk water-washing patents, is in regular use in many turbine oil systems.

Recently, progress has been made in reducing oxidation effects by the addition of anti-oxidants such as sulphur, nitro-cresol, mono-nitrobenzene, diphenylhydrazine, phenyl-α-naphthylamine, α-naphthylamine, etc. Certain oils also appear to contain anti-oxidants naturally. The role of the anti-oxidant appears to be to enter into the oxidation reaction itself, preventing, until it has been exhausted, any appreciable oxidation of the oil. This is a beneficial effect, so far as it goes, and it may become in this way possible to hold oxidation reactions down to minor or negligible limits in the future.

Tests on oil protected from oxidation by anti-oxidant agents, supplemented by water washing, are now in progress and will be reported in later publications.

THE INFLUENCE OF WATER ON THE LUBRICATING VALUE OF A COMMERCIAL MOTOR OIL*

By A. Fogg, M.Sc., and C. Jakeman †

The investigation was carried out to determine the maximum quantity of water which might be present in a lubricating oil of the type used for internal combustion engine lubrication without impairing the lubricating value.

Method of Drying the Oil. The method of drying the oil was suggested to the authors by the late Professor Browning. The sample of oil was placed in a Pyrex distillation flask, A (Fig. 1). The side branch of

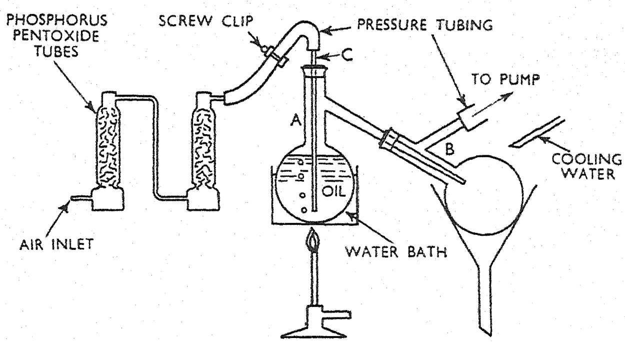


Fig. 1. Apparatus for Drying Samples of Oil

another Pyrex distillation flask B was connected to a "Hyvac" pump. A glass tube, C, drawn out at the end to a capillary, was inserted through the rubber stopper of flask A with the capillary reaching almost to the bottom of the flask. The other end of the tube was connected to two drying tubes containing phosphorus pentoxide (P_2O_5). Dry air was admitted to the oil through the tube C and the rate of flow controlled so that very small bubbles passed slowly through the oil for a few hours. The flask A was surrounded by a hot water bath, and flask B was cooled by a jet of water. All the joints were sealed with sealing wax dissolved in alcohol. A fairly high vacuum was maintained in the apparatus and the temperature of the water bath was controlled so as to prevent the oil from boiling. In order to avoid exposure of the

* Work carried out for the Lubrication Research Committee of the Department of Scientific and Industrial Research.

† Engineering Department, National Physical Laboratory.

dried oil to the air, the oil was stored in flask A until required for mixture with water.

Method of Mixing the Water with the Dried Oil. Samples were made up containing the following percentages (by weight) of distilled water: 0·02, 0·05, 0·1, 0·2, 0·5, 1·0. After tests on the first two of these samples it was found necessary to test samples containing smaller amounts of water and two further samples were made up containing 0·005 and 0·01 per cent. The Hurrell homogenizer used for this purpose consisted of a rotor ring and a stator ring, the distance apart of which could be varied from 0·003 to 0·015 inch. The speed of the homogenizer shaft was about 10,000 r.p.m. The oil was fed into the apparatus through a funnel and water was introduced to the funnel from a pipette, the quantities being kept proportional as far as possible. The rate of flow was regulated automatically by the machine, being dependent on the size of gap. The mixture was fed through several times with the smallest gap. To prevent heating of the oil, cold water was circulated through the body of the machine. Visual examination of the samples after passing through the machine showed no difference between the pure oil and the samples containing up to 0·05 per cent of water. The other samples appeared homogeneous, and after standing for a few days there was no sign of separation of the water from the oil in any of the samples. (After some months a light brown coloured sludge separated out at the bottom of the bottle in samples containing more than 0·1 per cent of water. No appreciable change in the colour of the oil was noticeable until 0·5 per cent of water was present. The two samples containing 0·5 and 1 per cent of water were brown in colour and this colour remained after the deposition of the sludge referred to above. It appears therefore that the sludge contained only a very small proportion of the water content.)

The results are tabulated against the known amount of water placed in the oil. It is possible that, during the unavoidable exposure of the oil to the air during test, an additional quantity of water may have been absorbed. No attempt has been made to measure the quantity of water in each sample after test, since the drying apparatus employed was not suitable for quantitative estimation of the water content.

Method of Test. The samples were tested in a National Physical Laboratory journal friction machine by determining the coefficient of friction at all temperatures from air temperature to seizure. The friction-temperature curve was taken as an indication of the quality of the oil, special attention being paid to the minimum value of the coefficient of friction and the temperature of seizure. Although great

care was taken in the preparation of the bushes, it was not always possible to repeat tests exactly, and in the present investigation the results obtained on the same bush were compared with each other. A bush was first run in with the dried oil until the friction-temperature curve was approximately the same on successive tests. A mixture of water and oil was then tried in the same bush and the alteration in the friction-temperature curve was examined. The only variable introduced was thus the quantity of water in the oil.

Conditions of the Tests. A nickel-chromium steel journal 2 inches in diameter was used with a bronze bush (88 per cent copper, 12 per cent tin, Brinell hardness No., 127), 2¼ inches long. The diametral clearance was 4 mils (0·004 inch), the load 1,000 lb. per sq. in. and the speed 1,300 r.p.m. (11·3 ft. per sec.). The sample of oil was pumped through the bearing and filtered in a wash-leather filter.

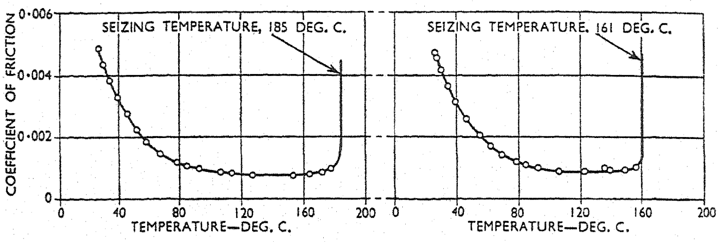
Tests on Bush No. 1 (Clearance 0·0042 inch). The dried oil was run in this bush until the seizing temperature was constant and was then replaced by a mixture of oil with 0·02 per cent of water. The observations are recorded in Table 1. The friction-temperature curves are given in Fig. 2.

TABLE 1. BUSH NO. 1

Lubricant	Time of running, hours	Coefficient of friction			Temperature of seizure, deg. C.
		At 40 deg. C.	At 70 deg. C.	Minimum	
Dried oil . .	0 – 7	$0{\cdot}0028_5$	$0{\cdot}0012_5$	$0{\cdot}0007_5$	161
	7 –13	0·0028	0·0013	$0{\cdot}0007_5$	168
	13 –19½	0·0030	$0{\cdot}0013_5$	$0{\cdot}0007_5$	169
	19½–22½	$0{\cdot}0030_5$	0·0014	$0{\cdot}0007_5$	180
	22½–25	0·0031	0·0014	$0{\cdot}0007_5$	186
	25 –29½	$0{\cdot}0031_5$	0·0014	$0{\cdot}0007_5$	185
Dried oil plus 0·02 per cent of water	0 – 1½	0·0030	0·0014	0·0008	160
	1½– 3½	0·0030	0·0014	0·0009	164
	3½– 5½	$0{\cdot}0030_5$	0·0014	0·0009	161

The addition of 0·02 per cent of water appeared to produce an increase in the minimum coefficient of friction and a decrease of 20 deg. C. in the seizing temperature. A visual examination of the surfaces of the bush and journal showed no appreciable change. They were lightly polished before a further test was made. During this test a violent seizure occurred at about 100 deg. C. and the surfaces were ruined.

This seizure may have been accidental, due to the presence of a small particle of dirt, and the tests were repeated on a new bush.



a *b*

Fig. 2. Temperature-Friction Curves

a Dry oil.
b Oil+0·02 per cent water.

Tests on Bush No. 2 (*Clearance 0·0040 inch*). The dried oil was run in this bush until the seizing temperature was fairly constant, after which the sample containing 0·02 per cent of water was tested. The test on the dried oil was then repeated. The sample containing 0·05 per cent of water was then tested, followed by a further test on the dried oil. The results of these tests are given in Table 2.

TABLE 2. BUSH NO. 2

Lubricant	Time of running, hours	Coefficient of friction			Temperature of seizure, deg. C.
		At 40 deg. C.	At 70 deg. C.	Minimum	
Dried oil . .	0 – 5½	0·0030	0·0013	0·0007	179
	5½–11½	$0{\cdot}0031_5$	$0{\cdot}0013_5$	0·0007	183
	11½–19½	0·0032	$0{\cdot}0013_5$	0·0007	198
	19½–24½	$0{\cdot}0032_5$	0·0014	0·0007	206
	24½–30½	$0{\cdot}0032_5$	0·0014	0·0007	203
Dried oil plus 0·02 per cent of water	0 – 4½	$0{\cdot}0032_5$	$0{\cdot}0013_5$	$0{\cdot}0008_5$	192
	4½–10	$0{\cdot}0032_5$	0·0014	$0{\cdot}0008_5$	189
Dried oil . .	0 – 6	$0{\cdot}0032_5$	0·0014	$0{\cdot}0008_5$	200
	6 – 9	0·0033	0·0014	$0{\cdot}0008_5$	199
Dried oil plus 0·05 per cent of water	0 – 1½	0·0032	0·0014	$0{\cdot}0010_5$	131
	1½– 3	0·0032	0·0014	0·0010	144
	3 – 5	0·0032	0·0014	0·0010	138
Dried oil . .	0 – 4	0·0033	0·0014	0·0010	159

These results show that 0·02 per cent of water had caused a rise in minimum coefficient of friction and fall in seizing temperature as before. After this test the dried oil approached its previous seizing temperature, but did not repeat the low value of $\mu_{\min}$, which suggests that the surface had been affected by the previous tests. On testing the sample containing 0·05 per cent of water, the effect on the minimum coefficient of friction and the seizing temperature was marked, while the machine ran unsteadily, showing variations in the friction moment. A fresh test of the dried oil showed that the surfaces had been spoiled and this was apparent upon visual examination. Tests were then made on a new bush with samples containing less water.

Tests on Bush No. 3 (Clearance 0·0039 inch). This bush was run in with the dried oil as before and samples of oil containing 0·005 per cent and 0·01 per cent of water were then tested. The test on the dried oil was then repeated. The results of these tests are given in Table 3.

TABLE 3. BUSH NO. 3

Lubricant	Time of running, hours	Coefficient of friction			Temperature of seizure, deg. C.
		At 40 deg. C.	At 70 deg. C.	Minimum	
Dried oil . .	0 – 3	$0{\cdot}0027_5$	$0{\cdot}0012_5$	$0{\cdot}0007_5$	—
	3 – 5½	0·0028	$0{\cdot}0012_5$	$0{\cdot}0007_5$	181
	5½– 7½	0·0029	0·0013	$0{\cdot}0007_5$	192
	7½–12½	0·0030	0·0013	0·0007	198
	12½–16	0·0030	0·0013	0·0007	199
Dried oil plus 0·005 per cent of water	0 – 2½	0·0030	0·0013	0·0007	200
	2½– 5	0·0030	0·0013	0·0007	197
Dried oil plus 0·01 per cent of water	0 – 5½	0·0030	0·0013	0·0007	197
	5½–10½	$0{\cdot}0030_5$	$0{\cdot}0013_5$	0·0007	196
Dried oil . .	0 – 4	$0{\cdot}0030_5$	$0{\cdot}0013_5$	0·0007	197

The results tabulated in Table 3 show that the seizing temperature and minimum friction are not changed by the addition of 0·005 or 0·01 per cent of water. The minimum quantity of water affecting the friction-temperature curve appears to lie between 0·01 and 0·02 per cent.

Static Friction of the Samples. The samples of dried oil and mixtures containing 0·005, 0·01, 0·02, and 0·05 per cent of water were

tested upon the Deeley oil-testing machine, using bronze pegs and a steel ring. These tests were made at air temperature (about 20 deg. C.), and although the observations were somewhat variable, the average result for all the samples was the same, namely, $\mu=0{\cdot}12$. This indicates that small admixtures of water which have a noticeable effect on the seizing temperature determined in the journal friction testing machine cannot be detected by the Deeley machine at air temperature.

Conclusions. (1) If a sample of the dry oil employed in these tests is intimately mixed with a little more than 0·01 per cent of water the seizing temperature is lowered and the minimum friction increased.

(2) 0·02 per cent or more of water in the oil deteriorates the condition of the running surface, the deterioration increasing rapidly with increase in the percentage of water under the conditions of the tests.

(3) Over the temperature range from air temperature to about 70 deg. C. the friction temperature curve of the oil is unaffected by the admission of water up to 0·05 per cent.

(4) There was no evidence that the water in these samples was removed by evaporation during the tests in the journal friction testing machine.

(5) Quantities of water up to 0·05 per cent cannot be detected by any increase in the coefficient of friction as determined in the Deeley testing machine.

It does not follow from these tests that the admission of a few drops of water in a sample of oil would produce the effect shown by these tests with intimate mixtures of oil and water. It is also probable that the water accumulated in an engine sump does not become so intimately mixed with the oil. Water in comparatively large drops may be more easily removed by evaporation and would affect the seizing temperature only when a drop of water happened to pass through the bearing during the critical stage—a matter of chance. There is some doubt as to the nature of the mixture of the water in an oil as supplied from the refinery. As a precautionary measure it is clear that according to these tests, the water content of an oil should be kept below 0·02 per cent.

MARINE ENGINE LUBRICATION PRACTICE

By Sterry B. Freeman, C.B.E., M.Eng., M.I.Mech.E.*

That the principles and design of the lubricating arrangements in marine engines are effective and are rightly carried out is made evident by the freedom from heated or excessively worn bearings and the concern caused when such do occur. That the oil industry has done its part in supplying oil of the right characteristics for the different uses involved is equally obvious. A number of marine superintendent engineers have been consulted and the general experience seems to be that they have had in recent years little serious trouble with lubricating oils. This is the more remarkable since the more modern types of machinery, the geared turbine and the oil engine, impose much heavier duties on the oil than the old reciprocating steam engines. The high temperatures in the superheated steam cylinder and the oil engine cylinder, the severe conditions in reduction gearing, and the long periods during which the oil is in use in a closed system are all more difficult conditions than obtained in general marine practice until after the War. Few of the old school of marine engineers were trained to appreciate abstract friction problems of power transmission, although they all rapidly acquired intimate experience of marine lubrication practice and realized that methods of application are as important as the quality of the oil. The majority accepted the oil purchased for their use with complete reliance on the supplier, and hoped for the best, but a more critical generation is now arriving, which wishes to know more about the lubricating oil and its suitability for the part of the engine for which it is intended.

Steam Reciprocating Engines. These engines are generally hand-oiled throughout except that in superheated steam engines the cylinder oil is introduced by means of a mechanical lubricator. If this is injected into the steam pipe, then the oil must be capable of being readily atomized. With hand-oiling there is often either a feast or a famine. It is the additional first cost of an enclosed engine and the difficulty of preventing the oil from entering the boilers via the piston rods which has prevented the adoption of the forced-lubricated reciprocating steam engine of any considerable sizes. In some cases, oil boxes and syphon feeding at various points of the engine are introduced and this makes for steadier lubrication. *Superheated steam* valves, pistons, and liners are lubricated by mechanical lubricators,

* Messrs. Alfred Holt and Company.

feeding into the main steam pipe by a special atomizing tube or to points around the high-pressure liner. In *saturated steam* jobs generally, cylinder oil swabbed on to the piston rods supplies all the lubrication required for the cylinders. The type of oil adopted for the bearings in the ships of the author's company complies with the specification set out in the Appendix, p. 478. This oil is, of course, unavoidably mixed with some water from the glands, etc., but this cannot be helped. It has been found from general practice that the following amounts of oil per 1,000 i.h.p. per day give good lubrication:—

Engines	Speed, r.p.m.	Working pressure, lb. per sq. in.	Steam temperature, deg. F.	Oil per 1,000 i.h.p. per 24 hr.	
				Cylinder, gal.	Engine, gal.
Cylinders, 31, 51½, and 86 inches dia.; stroke, 60 inches	75	190	360	$\frac{1}{10}$	2
Cylinders, 31½, 51½, and 86 inches dia.; stroke, 60 inches	75	190	520	$\frac{1}{5}$	2

Steam Turbines. A forced circulating system of lubrication is always adopted for lubricating turbine bearings, thrust blocks, and reduction gearings. The oil is either pumped directly to the bearings and the gearcase or to gravity tanks at the top of the engine room from which the oil runs at a constant head to the bearings and gearing sprayers. The gearcase drains to a tank constructed in the bottom of the ship to which the lubricating oil pumps are connected. The discharge pipe should be led below the normal working oil level in this tank to prevent aeration. It is advisable to arrange the suction and discharge pipes at opposite ends of the tank to give as long a path as possible between the two, thereby cooling the oil and depositing any impurities which may have been picked up in the system. A filter must also be installed in this system to prevent any small particles of metal or other impurities from being forced into the bearings.

The thrust block at the steam inlet end of the high-pressure turbine probably works under the worst conditions as the temperature there is fairly high. It is generally this bearing which gives trouble. There have been in the past few years upwards of fourteen cases in the author's company where the pivoted pads of the thrust block of the high-pressure turbine have become overheated and the metal has dragged and run. These thrust blocks present no unusual departure

from the standard design usual at the time these turbines were built (1919–26). The turbines are not fitted with clearance adjusting screws. Blocks with both spherical and flat adjusting blocks have failed in this way. The oil outflow pipes have been raised since the engines came out to ensure that the blocks are continuously flooded. The volume of the oil appeared ample and the temperature normal. No restriction of the flow of oil in the wells, oil supply belts, channels, and holes was apparent, and there was no dirt or obstruction. It was impossible to say whether each of the pads was taking the same load as the whitemetal was destroyed. In the same way the whitemetal lining of the spherical housing, where fitted, may have become worn at the root of the collar, allowing the fresh oil to bypass the thrust faces of the pads. The leading edge of the pads was well rounded and not faced with a sharp square corner that would scrape away the oil instead of allowing it to enter under the face of the pad.

The oil used for the average turbine installation is as shown in the specification set out in the Appendix.

Oil Engines. The arrangement for the return of bearing oil to the tank in the double bottom, and the course of the oil in that tank, apply to the oil engine as to the turbine. There is often a tendency to over-lubricate rather than to under-lubricate, especially in oil engine cylinders and compressors, and standard figures should be worked out for each type of engine. If these are exceeded an explanation should be required and found.

At the end of the compression stroke in the cylinder the piston rings are being forced out at a maximum pressure, say, 400–600 lb. per sq. in., against the liner wall; the gas temperature is at a maximum, say, 1,110–1,400 deg. C.; and the piston comes momentarily to rest. It is obvious that under these conditions boundary lubrication will be in force and that an ample supply of oil of a good load-carrying quality is needed. On the other hand, there is the danger that solids may build up in the piston ring grooves if too much or too unstable an oil is used; which points towards the use of a pure mineral oil with high stability to heat and oxidation, good oiliness and small frictional loss and high viscosity to effect sealing adequately.

The lubrication of piston and liner is carried out by mechanically operated lubricators delivering oil to two or more points on the cylinder liner in the scavenging area. The feed should be timed to inject the oil as the piston reaches the bottom dead centre after firing, and between the second and third rings from the top of piston. There should be one lubricator pump for each point of injection of each cylinder; the

number of points will vary with the size and type of engine. In oil engines where the cylinders open into the crankcase, the oil film formed on the cylinder walls is swept down into the crankcase, and all the oil is eventually exposed to heat sufficient to cause oxidation. Part of the oil is decomposed, resulting in formation of carbon, giving a black appearance. Separation and filtration remove the solids, leaving the oil darker than before, but the viscosity is not materially altered. With trunk engines a marked reduction of lubricating oil has been effected by fitting efficient skirts to the pistons. One of the worst causes of rapid wear is "blow past". The piston rings, if of the Ramsbottom type, should be located in their grooves in such a manner that exhaust gas must take a labyrinth passage to pass the piston. If the gaps in these rings come into line so that gas can blow past the rings, the oil film is destroyed and wear is rapid. The modern sealed rings obviate this defect. The piston ring wear is more serious in that the vertical wear makes the piston grooves deeper and ultimately ruins the initial uniformity of depth; a cast iron ring must then be fitted in the groove to save the piston.

Oil Engine Bearings. Shaft journals, pins, etc., of oil engines are lubricated by a forced circulation system, at a pressure of about 25 lb. per sq. in. The oil as a rule is led through the crankshaft, crank webs, and pins to all the bearings. The same lubricating oil is employed in many engines to cool the main pistons and is discharged to the same tank to be cooled, filtered, and returned to the circuit. During circulation under elevated temperatures, the contact with air, water, and metallic oxides tends to oxidize the oil, some of the resulting products being deposited where temperature is low and movement lessened. The effectiveness of oil coolers is lessened by sludge deposited in this way on the cooler tubes. The oiling system must be thoroughly clean before the engines are set to work. It is amazing how much rubbish comes out of the system when first scoured out by the oil.

In Diesel engine crankcase systems the oil should have the characteristics set out in the Appendix, p. 478. The sludge value clause was introduced into specifications for Diesel crankcase systems on the basis of work which had already been carried out on transformer oils. Cylinder bearing and compressor lubrication are separate problems and should be dealt with separately. There is an inclination to use one type of oil throughout, but it is doubtful whether the best results can be obtained in this way.

Mechanical Lubricators. These feed the exact amount required while the engine is running. Feed sights enable control to be kept of their

working and the amount being fed. The speed of the engine determines the amount and timing of the oil feed, and the supply of small quantities can be kept up with regularity, uninfluenced by temperature or other altering circumstance. Each point can be dealt with separately. These lubricators can pump against high pressures. It is a great help to have a small motor-driven oil pump and connexions fitted so that the oil reservoirs on these mechanical lubricators can be quickly and cleanly filled with the right oil without possibility of mistake or waste. Two-stroke engines have suffered from fires in the scavenge belts which in some cases have been traced to excess cylinder oil carried into these spaces by the scavenge air. In the hope of effecting a cure the amount of oil has been reduced, while ensuring that sufficient oil is injected to keep the piston rings in a free condition to prevent the passage of exhaust gases.

In the older type of engine the fuel is injected by means of blast air. This blast air is delivered by three-stage air compressors at a pressure of 800–1,200 lb. per sq. in. These air compressors need special lubrication which acts as a piston seal and protects the cylinder walls. This is effected by means of mechanical lubricators of similar type to those used for the main engine cylinders. As there is always a certain amount of water vapour present in the compressors, a saponifying type of oil must be used, as set out in the Appendix.

Condition and Cleaning. The full flow of oil is passed through strainers, and a portion, say about 2 per cent, is bypassed into a centrifugal separator, which is started as soon as the oil system pumps and the clean oil is returned to the system. The correct size of diaphragm should be used to make certain of completely purifying the oil passed. For good separation a temperature of about 180 deg. F. is desirable.

In a typical installation of a six-cylinder two-stroke supercharged double-acting engine and three three-cylinder auxiliary engines, the separator passed 110 gal. per hr. and extracted on an average 0·057 per cent of sludge. This was sufficient to keep the system in a sweet condition. Make-up oil amounting to about 20 per cent of the bulk is introduced every four to five months, so that 100 per cent is renewed in, say, two years' time. Any additions of new oil should be made gradually, as there is a tendency to form sludge if a large amount of new oil is introduced into the system too rapidly. Oil which has become more or less contaminated in use is dealt with by a settling tank with an internal steam coil. The bulk of the water and heavier material is thrown down and the oil is drawn through a high suction valve to a heater, whence it passes through a centrifugal separator back

TABLE 1. OIL SUPPLIES FOR MARINE ENGINE LUBRICATION

Dimensions			S.h.p.	Speed, r.p.m.	Type	Remarks	Oil per 1,000 h.p. per day, gal.			Total oil per 1,000 h.p. per day, gal.
No. of Cylinders	Bore, mm.	Stroke, mm.					Engine	Cylinder	Compressor	
Main Engines:—										
8	740	1,150	3,000	115	Four-stroke single-acting	Unsupercharged air injection	1·84	1·08	0·19	3·11
8	630	1,100	1,850	125	Do.	Do.	1·94	0·99	0·11	3·04
6	620	1,300	2,750	140	Do.	Supercharged air injection	0·77	1·08	0·05	1·90
8	740	1,500	4,300	110	Do.	Do. Airless injection	0·86	0·89	—	1·75
6	450	1,200	3,300	120	Two-stroke double-acting	Do. Do.	0·48	1·25	—	1·73
Auxiliary Engines:—										
3	320	350	160	300	Four-stroke single-acting	Air injection	8·0	3·4	3·2	14·6
3	12·8	16·0	160	300	Do. Do.	—	3·9	0·80	0·80	5·5
3	330	600	250	300	Four-stroke single-acting	Airless injection	7·3	*	1·2	8·5
4	220	370	250	400	Two-stroke single-acting	Airless injection	17·7	0·89	0·41	19·0

* Compressor oil used for cylinders to prevent wrong oil being used on compressor.

to the oil sump. Unfortunately these separators can only extract matter with a different specific gravity from that of the oil. When going through the centrifugal separator it has been found good practice to allow a small amount of hot water at a temperature of 180–200 deg. F. to mix with the oil before it is separated. Some oil will be lost in the form of sludge from the resulting emulsion, but once the oil has all been passed through the washing process the consumption will be restored to normal. The broad fact is now realized that lubricating oil cannot be injured by mechanical use, and the old belief that oil "loses its nature" through punishment of this kind is disappearing. Loss in lubricating value is caused by impurities formed in or picked up by the oil and the only reason for which oil is now condemned as unfit for use is that the contamination by salt water or fuel oil is so great that it would be uneconomical to attempt to purify it.

The services of an oil specialist to deal with the physical and chemical qualities of the oil purchased are essential if high lubricating efficiency is to be obtained and maintained. His work falls into three classes: (1) the examination of new oils, (2) the examination of the oil in use in systems, and (3) post mortems.

The tests of new oils are intended to cover general quality and to decide whether or not satisfactory service can be expected in specific cases. The examination of oils actually in systems is made to ensure that such oil is in a satisfactory condition, that its breakdown by oxidation in service is not taking place at a faster rate than is made up by fresh supplies, and to keep a careful eye on dilution by fuel or water, sea or fresh. The third class—post mortems—explains itself. Such extraordinary happenings have occurred as the accidental introduction of linseed oil and of bean oil into lubricating systems, leading to curious results and lengthy and costly periods of elimination.

Lubricants. In practice the limits between which oils must lie to give efficient lubricating service in any given type of engine are wider than were formerly believed. The principle governing lubrication is that suitable oil must be available at the points in the machinery where it is wanted, when it is wanted. Heavy sludge in the crankcase oil may ultimately result in the oil not arriving at the lubrication points when it is required, and mechanical breakdown is the result. If sludge is not removed its presence leads to the production of more sludge on a sort of compound interest law; hence the advisability of continuous separation of the oil under suitable conditions. Provided that the oils have the same broad general characteristics, no bad results will be found in practice on mixing oils from different suppliers.

After examining a large number of samples of various lubricating

oils, it is difficult for the purchaser to understand the difference in the prices of these oils. It would be of much interest if those who control the supply and sale of lubricating oil would indicate whether, in their view, we are faced with a shortage of supplies in this area of oil production. In view of the fact that oil is not easily destroyed, although it may become contaminated, has the time arrived to add largely to our resources in the matter of oil saving and reconditioning plant ? The amount of oil that is allowed to run away after being used once must be considerable, and much of it cannot be greatly the worse for having passed through a bearing before being lost.

APPENDIX

Lubricating Oil Specifications

Steam Reciprocating Engine Cylinder Oil :—

Flash point, above 500 deg. F.
Specific gravity, 0·908–0.912.
Viscosity (Redwood) at 140 deg. F., 950–970 sec.
,, ,, ,, 200 deg. F., 215–225 ,,

Steam Reciprocating Engine Oil. To consist of a blend of 87·5 per cent by volume of refined red mineral oil and 12·5 per cent by volume, of genuine East Indian rape oil (blown). The blend to show the following viscosities (Redwood) :—

at 70 deg. F., 1,900–2,600 sec.
,, 140 deg. F., 205– 235 ,,
,, 180 deg. F., 90– 100 ,,

The rape oil to be blown to a viscosity which will give the above figures in the blended oil.

No separation must be observable in the blend after it has stood for 24 hours at a temperature of 25–30 deg. F. Buyers reserve the right to draw samples of the constituent oils at works.

Turbine Oil. To conform to the following:—

Flash point (Pensky Martens), above 400 deg. F.
Viscosity (Redwood) at 70 deg. F., 1,400–1,800 sec.
,, ,, ,, 100 deg. F., 450– 550 ,,
,, ,, ,, 140 deg. F., 150– 170 ,,
,, ,, ,, 180 deg. F., 65– 75 ,,

To ensure rapid separation from water the demulsibility by the

standard test of the Institution of Petroleum Technologists (I.P.T.) is not to exceed 3.

Steam Auxiliary Engine Lubrication Oil. To have:—

Flash point (Pensky-Martens), not below 375 deg. F.

Viscosity	(Redwood)	at 70 deg. F.,	about	1,400	sec.
,,	,,	,, 100 deg. F.,	,,	475	,,
,,	,,	,, 140 deg. F.,	,,	160	,,
,,	,,	,, 180 deg. F.,	,,	75	,,

To give rapid separation from water.

Diesel Engine Crankcase Oil. To have the following characteristics:—

Flash point (Pensky Martens), above 400 deg. F.

Viscosity	(Redwood)	at 70 deg. F.,	1,600–2,000	sec.
,,	,,	,, 100 deg. F.,	525– 625	,,
,,	,,	,, 140 deg. F.,	170– 185	,,
,,	,,	,, 180 deg. F.,	80– 85	,,

The sludging value (Michie) as determined by the standard I.P.T. test is not to exceed 0·8 per cent.

Diesel Engine Air-Compressor Oil. To have:—

Flash point (Pensky Martens), above 400 deg. F.

Viscosity	(Redwood)	at 70 deg. F.,	1,800–2,200	sec.
,,	,,	,, 100 deg. F.,	575– 675	,,
,,	,,	,, 140 deg. F.,	180– 200	,,
,,	,,	,, 180 deg. F.,	85– 90	,,

To be slightly compounded (say, 3 per cent) with neutral fatty oil.

Diesel Engine Cylinder Oil. To have approximately the following characteristics:—

Specific gravity at 60 deg. F., 0·897.

Flash point (Pensky-Martens), above 500 deg. F.

Viscosity	(Redwood)	at 140 deg. F.,	575	sec.
,,	,,	,, 180 deg. F.,	220	,,

LUBRICATION UNDER HIGH-TEMPERATURE CONDITIONS WITH REFERENCE TO THE USE OF COLLOIDAL GRAPHITE.

By H. Higinbotham, B.Eng.*

Temperature is a prominent factor in cylinder lubrication, whether the engine is driven by petrol, oil, or steam. If its magnitude varies in different types of engines, it is also a limiting factor in mechanical design. Nägel (1929) has shown, quoting figures of Eichelberg, that the amount of heat flowing through the oil film on the cylinder walls and piston of a low-speed two-stroke Diesel engine varies cyclically from approximately 30,000 B.Th.U. per sq. ft. per hr. to 400,000 B.Th.U. per sq. ft. per hr. After five minutes' working up from cold the temperature at the surface of the cylinder walls in question is 424 deg. F. This figure indicates a lower general temperature factor in engines of this type, for the temperature on the explosion side of the oil film will be considerably higher.

Teetor (1936) reports a wide difference in the temperature of the cylinder walls of a water-cooled petrol engine, according to the location, his figures varying from 490 deg. F. to 260 deg. F. at a position near the top dead centre on the valve side and on the opposite side to the valves, respectively, for an engine speed of 2,000 r.p.m., and at 3,500 r.p.m. these figures become, respectively, 530 deg. F. and 285 deg. F. Of particular interest is his statement that when the surface temperature in the cylinder exceeds 400 deg. F. a breakdown in lubrication may be expected. He also indicates that piston scuffing implies an instantaneous temperature of the metal of about 1,000 deg. F.

The thermal conditions in engines working on saturated steam are less rigorous than in those working on petrol (with explosion temperature of the order of 4,000 deg. F.) or in oil engines. Temperatures of 250 deg. F. to 300 deg. F. on the surface of the cylinder wall, for an engine operating on steam at 117 lb. per sq. in., may be taken as typical. For a variation of 200 deg. F. in steam temperature, the cylinder wall at its hottest part would be at about 300 deg. F. In engines working on superheated steam the thermal factor is comparable to that in internal combustion and compression-ignition engines. Steam temperatures of 600 deg. F. and more are, of course, not uncommon.

In engines operating on blast furnace gas, temperatures on the cylinder walls would be around 300 deg. F. while information in the possession of the author suggests that the piston rings in such an engine

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may attain red heat, due to the high temperature of the incoming gas. The explosion temperature would, however, be lower than in petrol engines.

This brief summary of thermal conditions shows that a lubricating oil, where used, must be working near its critical temperature. At certain parts of the heat cycle conditions may in certain engines exceed this temperature. In four-stroke and two-stroke engines the cylinder walls are exposed at the end of the power stroke. Being heated to their maximum temperature it is during the ensuing exhaust stroke that the thermal factor in lubrication is most felt. In the two-stroke type the incidence of maximum pressure with maximum explosion temperature makes lubrication difficult. Piston and ring pressure and the reciprocating motion of the piston are conditions closely associated with temperature in promoting cylinder wear.

Colloidal graphite is essentially a high-temperature lubricant, as graphite will resist oxidation up to a temperature of about 1,000 deg. F., whereas all lubricating oils would oxidize at that temperature so rapidly as to ignite spontaneously. Graphite is very inert, and this is an important quality since the rate of decomposition of an oil and its reactivity above 650 deg. F. increase rapidly with small increments in temperature. Colloidal graphite can, in consequence, give support to lubrication above this temperature. Boundary conditions of lubrication will predominate at the top of the cylinder in internal combustion engines, and at times dry friction may occur, so that it is essential that colloidal graphite of the highest purity and fineness with respect to particle size is employed. Specification of the product used is, for this reason, necessary.

Values for friction under working conditions in the cylinder are difficult to obtain, but as a basis for comparison it may be stated that the coefficient of friction of a journal bearing surface at 500 r.p.m. and 230 lb. per sq. in., lubricated with a full fluid film of medium viscosity oil, would be of the order of 0·001 at a temperature of about 100 deg. C. and over. Under boundary conditions this would be approximately 0·01 to 0·10, while the figure for a boundary layer of oil plus an adsorbed surface of graphite would be of the order of 0·005 to 0·01. Dry friction of metal on metal gives a coefficient of from 0·1 to 0·3 according to the metals concerned. The formation of an adsorbed graphoid surface thus provides an effective lubricating barrier between fluid conditions, boundary conditions, and metallic friction. The coefficient of friction of a bearing under fluid film conditions of lubrication is not sensibly affected by the presence of colloidal graphite. There is a tendency, however, in practice towards a slightly lower average value.

The graphoid surface formed on a bearing face lubricated with oil containing colloidal graphite is composed of tile-like particles of the material lying parallel to the underlying metal. It is an adsorbed surface of a thinness that precludes direct optical measurement. Mabery (1913) first indicated its presence by friction experiments, while Jenkins (1934) and latterly Finch, Quarrell, and Wilman (1935) have supplied confirmation by electron diffraction methods and determined its structure.

The electron diffraction pattern of a graphoid surface on a steel face lubricated with oil containing colloidal graphite is reproduced in Fig. 1. The diffuse spots lying on the concentric arcs at their apexes indicate that the particles lie flat like tiles. This position is necessary if mechanical protection of a bearing face, such as a cylinder wall, is to be afforded against corrosion, abrasion, and thermal oxidation, and the maximum lubricating value of the graphite is to be obtained. Another surface formed from colloidal graphite is shown in Fig. 2, the pronounced diffraction spots emphasizing the homogeneous disposition of the particles. Using X-ray diffraction methods J. J. Trillat (1937) has confirmed, using the same material, the results obtained with electron diffraction in England. His X-ray spectrum is shown in Fig. 3, the basal (002) orientation of the particles being clearly revealed by the line for that plane. He has also shown that the polar groups of a fatty body are encouraged to orientate on a surface treated with colloidal graphite, the X-ray pattern for the polar groups on plain polished iron being given in Fig. 4 and for the molecules on the treated surface in Fig. 5.

Tests carried out at the National Physical Laboratory (circa 1928) indicate that colloidal graphite increases the seizing temperature of an oil, permits the oil film to withstand higher bearing pressures, enables the feed of lubricant to be reduced, and minimizes the possibility of seizure. Where seizure is imminent in a bearing, the temperature will rise to a high value, and often an oil will be carbonized owing to the high localized temperature. The fact that colloidal graphite delays seizure under these conditions shows that lubrication is maintained at least temporarily. The use of this adjunct lubricant for the lubrication of kiln and annealing-car bearings, and of oven chains is another indication of its value at high temperatures.

There is thus evidence to show that colloidal graphite is an effective lubricant for the cylinder zone. In internal combustion engines, it may be passed into the cylinder by the use of a graphited crankcase oil. It may also be added to the petrol, when the minute particles of graphite are drawn in by the induction stroke and swept on to the friction faces, the inlet valve stem being so lubricated. In both cases a reduction of

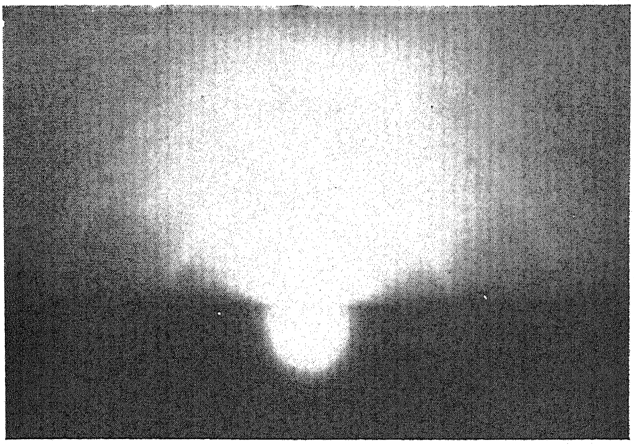

Fig. 1. Electron Diffraction Pattern of a Graphoid Surface on Steel

The diffraction spots indicate the presence of orientated graphite particles (Finch).

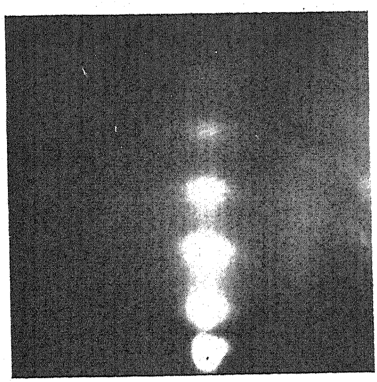

Fig. 2. Electron Diffraction Pattern of Colloidal Graphite Particles lying with Typical Basal Orientation on a Substrate and Completely Covering it (Finch)

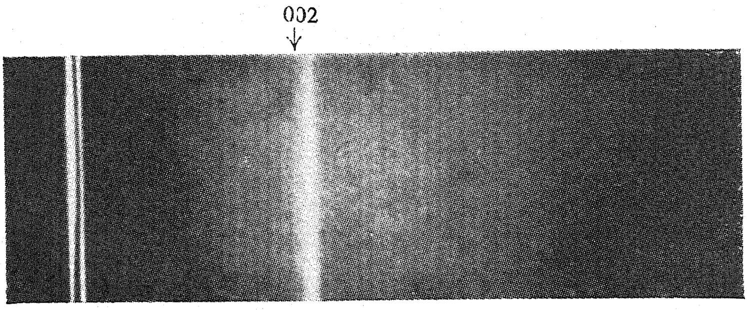

Fig. 3. X-ray Spectrum of Particles of Colloidal Graphite which lie on their Substrate with a Perfect Basal Orientation, obtained without Friction (Trillat)

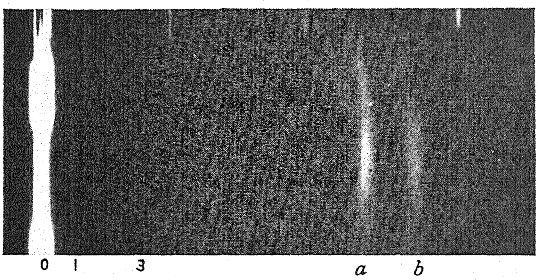

Fig. 4. Stearic Acid (Fused) on Polished Iron (Trillat)

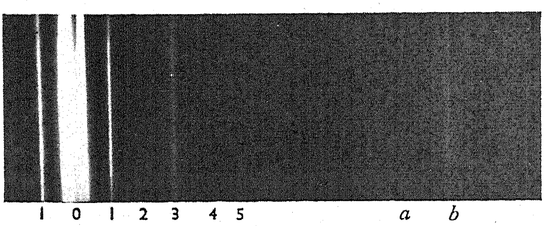

Fig. 5. Stearic Acid (Fused) on Polished Iron Treated with Colloidal Graphite (Trillat)

wear has been reported in new and run-in engines. A test carried out by the Research Department of the Institution of Automobile Engineers has shown that sump oil containing colloidal graphite reduces cylinder wear by one half in a new engine in which repeated cold starts were made (Fig. 6). While the major factor in the test contributing to wear might have been other than high temperature, the value of colloidal graphite at cylinder temperature could be inferred.

Further tests carried out by the Research Department of the Institution of Automobile Engineers with colloidal graphite added to the fuel of a water-cooled engine operating under similar conditions to

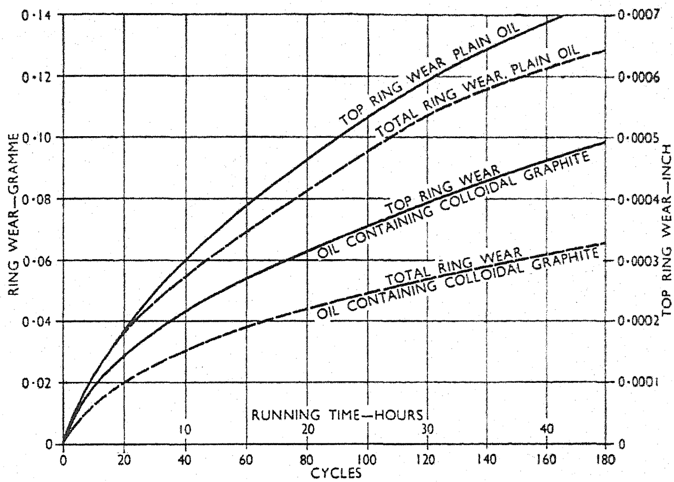

Fig. 6. Effect of Colloidal Graphite on Piston Ring Wear

that mentioned above, also indicate that a protective action is exerted by colloidal graphite. Stuart has measured the power output from a small petrol engine operating with plain petrol and petrol containing colloidal graphite, his results being shown in Fig. 7. The absence of a dip in the power curve with graphited fuel is to be noted.

While bench tests may be open to criticisms, figures for wear have been obtained on large fleets of vehicles, the following being an example: A fleet of tank wagons in 1934 gave an average mileage before reboring, considered to be necessary at 0·015 inches of wear, of 46,000 miles. The use of petrol containing colloidal graphite was adopted in 1935, and the mileage increased to 58,000 miles. The adoption of graphited crankcase oil, in addition to graphited petrol, in 1936 brought the mileage up to 77,000 miles.

Wolf (1934b) states that colloidal graphite reduced cylinder wear in a marine Diesel engine. On two successive voyages to the East the average liner wear reached the very high value of 0·015 inch and 0·012 inch, respectively. When one of the cylinders of the engine was lubricated with the same oil, but containing a small percentage of colloidal graphite, wear was reduced to 0·008 inch, and ring breakage was eliminated. Improved finish of the walls of the liner and rings may be expected when using oil containing colloidal graphite for lubrication.

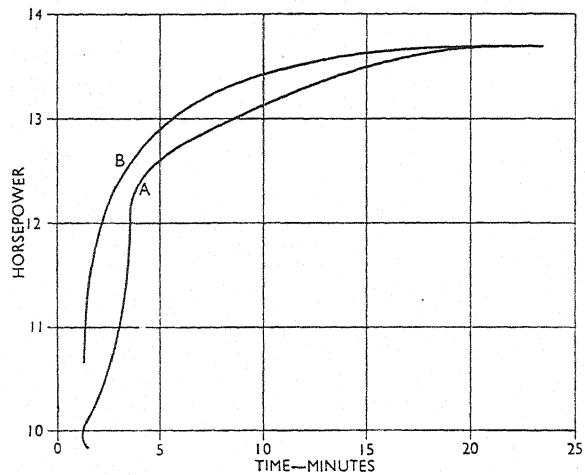

Fig. 7. Power Output from a small Internal Combustion Engine Starting with Plain Petrol (A), and Petrol containing Colloidal Graphite (B) (Stuart)

A feature in large stationary Diesel engines is the area of friction faces exposed to high temperature in the cylinder. Furthermore, the heat factor is found to be more severe in this type of engine, as opposed to petrol engines, because the rise in temperature during the power stroke is slower and takes place concurrently with the exposure of the wall of the piston in its downward stroke. Hillman records temperatures of 300 deg. F. on the surface of the second piston ring from the top, and 230 deg. F. on the bottom-ring. The gas temperature is, of course, far higher; Horiak (1933) quotes values for the gas in a high-speed Diesel engine which vary cyclically from about 3,600 deg. F. to 4,860 deg. F., according to the injection period. It is largely due to

the lower piston speeds and the more evenly distributed pressure on the liner that wear in Diesel engines is not greater than it is. The necessity for efficient lubrication at high temperature, however, remains.

Compressors present their own problems of lubrication, but it may be mentioned that colloidal graphite in an oil permits a lower feed while maintaining stable lubrication, so that the possibility of spontaneous ignition of oil due to excess lubricant can be minimized.

Steam Engines. Colloidal graphite is proving of considerable value in high-speed steam engines. Certain manufacturers are fitting high-speed engines with lubricators which feed in continuously a dispersion of colloidal graphite in water. As with the use of an oil dispersion, a graphoid surface is formed on the cylinder walls and piston rings, which produces a mirror-like finish. Bright spots, associated with inadequate cylinder lubrication, are noticeably absent with this form of lubrication. It is reported that the exhaust steam can be used for process work, while condensate may be returned to the boiler without the possibility of priming. A characteristic trouble sometimes met with in steam cylinders is groaning during starting up and it is reported that this defect is eliminated by colloidal graphite.

Wolf (1934a) suggests the use in vertical steam engines of 0·15 to 0·6 gramme of water containing 0·33 per cent of colloidal graphite per b.h.p. per hr. for engines below 400 h.p., and 0·05 to 0·4 gramme for engines above 400 h.p. The consumption for marine propulsion units would approach the lower figure. It has been found that for high-speed engines working on saturated steam the feed of a dispersion of colloidal graphite in water to the cylinders necessary for adequate lubrication varies according to local conditions from 2 drops per minute to 6 drops per minute. Johnston (1916) described the use of colloidal graphite over four years with saturated steam at 120 lb. per sq. in. A six months' running test on high-speed engines gave cylinder wear of only 0·001 inch, and the author states that the high-pressure cylinder and piston rings were in faultless condition, having mirror-like surfaces. The importance of using a fine dispersion of graphite must be emphasized, reference being made to the work of Finch and others (1935), who sum up the difference between colloidal and ordinary powdered graphite by saying that films "prepared from the colloid are better lubricants and more easily wetted by oil, have greater covering capacity . . . show greater powers of adherence than those formed from a graphite suspension."

For engines operating on superheated steam colloidal graphite dispersed in oil or in water may be used. Data on this application are being collected.

REFERENCES

FINCH, QUARRELL, and WILMAN. 1935 Trans. Faraday Soc., vol. 31.
HILLMAN. "Elements of Diesel Engineering", by O. Adams.
HORIAK. 1933 *Gas and Oil Power*, vol. 28, p. 256.
JENKINS. 1934 *Philosophical Magazine*, 7th Series, vol. 17, p. 457.
JOHNSTON. 1916 Birmingham Association of Mech. Eng.
MABERY. 1913 *Journal of Industrial and Engineering Chemistry*, vol. 5, p. 717.
NÄGEL. 1929 *Engineering* (various issues), "Transfer of Heat in Reciprocating Engines".
STUART. "Upper Cylinder Lubrication—Its Relation to Cylinder Wear", published by E. G. Acheson, Ltd.
TEETOR. 1936 S.A.E. Journal, vol. 39, p. 328.
TRILLAT, J. J. 1937 *Journal de Chimie*, vol. 34.
WOLF. 1934a Trans. Inst. Mar. Eng., vol. 46, part 5, p. 169.
1934b Trans. Inst. Mar. Eng., vol. 46, part 7, p. 197.

LUBRICATING PRACTICE ON THE CANADIAN NATIONAL RAILWAYS

By S. J. Hungerford*

In the operation of locomotives and cars in Canada, atmospheric temperature is of particular importance and must always be kept in mind when planning the composition and physical properties of the lubricants. As an example, in the middle of winter, along the three thousand mile stretch from coast to coast, the temperature may range from 50 deg. F. or more above zero to 30 or 40 deg. F. below zero, and some grades of oil or of grease which would give excellent service at the higher temperature would be totally unfit for use in severe cold.

As a result of these exacting conditions, very careful study has been given to the subject and thorough service trials have been made with various types of lubricants, and by means of these practical tests, certain properties were found unsatisfactory and were eliminated, while others, which gave good results, were retained and adopted as standard.

This brief statement will serve to give a general explanation why certain properties have been adopted, and it will be understood that under warmer or more uniform conditions of temperature, other properties or types of lubricants might be preferred.

The principal lubricants used on the locomotives and cars are as follows:—

Car Oil. It has been found impracticable to use a "summer oil" and a "winter oil" in journal boxes, and for some years a single "year-round" car oil has been used. In this oil, two extremes have been established:—

(1) The oil must have a viscosity of at least 45 sec. Saybolt at 210 deg. F., since this viscosity, after careful investigation, was found to be near the minimum with which good service results could be obtained in cars operated in the southern United States.

(2) The oil must remain fluid as long as possible when subjected to cold, and after being frozen it must become fluid at a low temperature, approximately −35 deg. F.

In order to determine what may be expected during cold sub-zero weather, the oil is first frozen, using "dry-ice" and acetone; the container holding the oil is then removed from the freezing mixture and stirred slowly with a cold test thermometer, the temperature being noted at which there is a noticeable movement of the oil within five

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seconds when the container is turned upside down. This temperature is termed the "cold test" of the oil.

The pour point method of the American Society for Testing Materials, with preliminary preheating to 115 deg. F., has been found unsatisfactory and is not used, since with certain types of oils the results proved very misleading and did not indicate what occurred in service.

The car oil which is used all the year round for both passenger and freight cars and upon locomotives has roughly the following properties:—

Specific gravity at 60 deg. F.	24·8 deg. American Petroleum Institute.
Flash point (open cup)	355 deg. F.
Viscosity, Saybolt universal:—	
at 100 deg. F.	200 sec.
at 210 deg. F.	45 sec. minimum.
−30 deg. F., A.S.T.M. chart	200,000 ,,
Precipitation No.	below 0·05
Cold test	−35 deg. F.

This oil gives generally good service, and the journal boxes are packed with "dope" prepared by saturating with the oil a good grade of wool waste purchased under specifications and tested prior to acceptance.

Wicks or other feeding devices cannot be used owing to the climatic conditions outlined above.

Valve Oil. One grade, termed "Superheat Valve Oil," is used on road engines, and as the cold test of a rather heavy-bodied oil of this type is necessarily rather high, special arrangements are made to protect the feed lines from cold so as to ensure a steady flow of oil to and from the lubricators under all conditions of temperature and weather. The properties of this oil are as follows:—

Specific gravity at 60 deg. F.	25·3 deg. American Petroleum Institute.
Flash point (open cup)	585 deg. F.
Viscosity, Saybolt universal, at 210 deg. F.	160–200 sec.
Fat (tallow and wool fat)	4 per cent
Precipitation No.	less than 0·05
Cold test (without preheating):—	
Pours at	45 deg. F.
Workable at	+10 deg. F.
Hard at	0 deg. F.

Driving Axle and Crankpin Grease (*Grease No.* 1). One grade is used for both purposes, the composition and properties being about as follows:—

Mineral oil (valve oil)	53 per cent by weight
Soap (tallow or wool fat base)	46 ,, ,,
Consistency, A.S.T.M. cone, 70 deg. F.	34
Softening point (Schroeder)	212 deg. F.

The aim is to maintain such consistency that the grease shall not be soft enough to be thrown off from rods or pins at operating temperatures and speeds of the parts, and on the other hand the grease must not be unnecessarily stiff.

Grease No. 4. Engine truck and main driving journals equipped with floating bush boxes, also crosshead wrist pins.

Mineral oil	62 per cent by weight
Soda soap	37·5 ,, ,,
Moisture	0·4 ,, ,,
Consistency:	
Penetration, A.S.T.M. cone . .	67
Softening point (Schroeder) . .	220 deg. F.

Valve Motion Grease (*Grease No.* 8). Motion and driving wheel hub faces.

Loss at 212 deg. F.	0·4 per cent by weight
Mineral oil	84·7 ,, ,,
Soda soap	14·5 ,, ,,
Consistency:	
Penetration, A.S.T.M. cone . .	223
Melting point	288 deg. F.

Stoker Gear Case Grease (*Grease No.* 9).

Mineral oil	99 per cent by weight
Soda soap	1 ,, ,,
Consistency	Flows at 70 deg. F. like stiff molasses.

Air Pump Oil—Air Cylinders.

Specific gravity at 60 deg. F. . . .	25·0 deg. American Petroleum Institute.
Flash (open cup)	450 deg. F.
Viscosity, Saybolt universal:—	
at 100 deg. F.	900 sec.
at 210 deg. F.	79 ,,
Emulsion test	Good separation

Air Brake Lubricant. In composition it should consist of mineral oil and lime or soda soap.

Mineral Oil.	
Viscosity at 100 deg. F., Saybolt universal	not less than 100 sec.
Cold test	not above 0 deg. F.
Grease.	
Melting point	not less than 160 deg. F.

The consistency should be as follows:—

(*a*) At 70 deg. F. the consistency shall be such that a spindle, one inch in diameter and of a total weight of 90 grammes, when placed

lightly upon the surface of the grease and released shall sink approximately one inch into the grease in 15 seconds.

(*b*) At -30 deg. F. the consistency shall be such that a glass rod of $\frac{1}{4}$ inch diameter, weighted with 5 lb., shall penetrate at least one-half inch in 15 seconds and can be withdrawn with little effort.

Special Anti-Friction Bearing. Both oil and grease are used and the recommendations of manufacturers are followed.

AERO-ENGINE LUBRICATION AND THE PERFORMANCE OF MODERN OILS

By O. T. Jones, B.Sc.,* and E. E. Turner†

Progress in the design and manufacture of aircraft engines proceeds at a remarkable rate. This progress shows itself in engines of higher horse-power, reduced weights per brake horse-power, higher piston speeds, and higher mean effective pressures. As a result, the lubrication of aircraft engines receives the constant study of engine designers and lubricating oil producers.

Foord's statement (1929) that "Lubrication is the heart of an engine" applies in the strongest possible terms to aircraft engines. Since then much progress has been made in the refining of aircraft lubricants, but engine progress has been so rapid that the refiner is constantly exercised in his desire to keep ahead of the lubrication requirements of modern engines. Designers of world-wide repute (Fedden 1934, and Wilkinson 1933–4) have stressed the importance of improved lubricants in order to allow wider scope for further development of engine performance. Foord (1929) has outlined the ideal characteristics of an aircraft engine lubricant, stressing the importance of oil fluidity at zero deg. C., viscosity temperature curve, carbonization, etc., laying particular stress on the importance of internal cleanliness of engines after endurance tests, as regards deposit formation in the critical parts of oil-circulating systems and around piston rings.

The greater power outputs achieved in recent years are associated with higher piston speeds, and oil temperatures, heavier bearing pressures, increased rubbing speeds, and higher gas pressures and temperatures, all of which have a marked influence on the performance of lubricating oil. As instancing the progress made in air-cooled engines, the following is quoted from the *Bristol Review* (1935) regarding Bristol engines of the same cylinder capacity: "Over this period of 15 years the output per litre of cylinder capacity has been increased by 130 per cent; the specific weight in spite of the addition of gearing and supercharger, has been reduced by 40 per cent; brake mean effective pressures have increased from 112 to 185 lb. per sq. in.; fuel consumption has been reduced by 25 per cent; maximum crankshaft speeds have increased from 1,625 to 2,925 r.p.m. . . ."

As regards liquid-cooled engines, Wood (1936) mentions Rolls-Royce "Kestrel" engines XIV–XVI, having maximum ratings of

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745 h.p. at 3,000 r.p.m. at 14,500 feet altitude, a dry weight of 955 lb., a maximum piston speed of 2,750 ft. per min. with a brake mean effective pressure at maximum engine speed of 151·5 lb. per sq. in., giving 35·1 h.p. per litre of cylinder capacity. He also quotes considerably higher brake horse-powers for special engines. For liquid-cooled engines using ethylene glycol as cooling medium, outgoing oil temperatures, normal and maximum, are quoted as 110 and 130 deg. C. respectively.

Wilkinson (1933–4) has quoted early War-period aircraft engines which varied from 3 to nearly 5 lb. per h.p., together with interesting data on engine development from 1915 to 1933, also showing improvements in weight and fuel consumption. It is interesting to note that Wilkinson (1934) has outlined the advantages of aero-engines with small-capacity cylinders operating at a high number of cycles per minute. Modern tendencies in engine design are of special interest in relation to these claims.

The foregoing gives a brief indication of the performance of aircraft engines up to 1935–6, since when further development has been made.

Lubricating oils for aircraft engines have also developed rapidly. In 1929 motor car oils of suitable viscosity were tested and approved by certain aero-engine builders owing to their universal availability overseas, as existing aircraft engine lubricants were not widely distributed. One such lubricant had the following characteristics when compared with the then Air Ministry requirements:—

	Motor oil	"P. 4" specification
Specific gravity at 60 deg. F.	0·911	—
Viscosity, poises at 100 deg. F.	2·38	Max. 2·9
" " " 200 deg. F.	0·179	0·183 to 0·207
Open flash point, deg. C.	250	Min. 199
Cold test at 0 deg. C.	Does not flow	Min. 1
Coke value, per cent	1·29	Max. 0·65
Oxidation (viscosity) ratio	2·19	Max. 2
Ash, per cent	Negligible	Max. 0·25

This oil satisfactorily completed a 100-hour test in a Bristol "Jupiter VIII" engine, running at 2,000 r.p.m. and developing 380 b.h.p. The outgoing and ingoing oil temperatures were approximately 92 and 70 deg. C. respectively.

A modern aircraft engine oil has the following characteristics, which demonstrate the improvements effected:—

	Modern aero-engine oil	Air Ministry specification D.T.D. 109
Specific gravity at 15 deg. C.	0·882	—
Viscosity, poises at 100 deg. F.	2·49	Max. 2·9
,, ,, ,, 200 deg. F.	0·203	Min. 0·183
Open flash point, deg. C.	282	Min. 199
Coke value, per cent	0·18	Max. 0·65
Coke value after blowing	0·63	Max. original +1·0
Oxidation (viscosity) ratio	1·38	Max. 2·0

This oil completed in 1935 a 100-hour test in a Bristol "Pegasus III M.3" engine with the following results: 800 b.h.p. at 2,300 r.p.m. run in 10-hour periods. The engine was started up from cold, using oil at atmospheric temperature at the commencement of each 10-hour period. Throttle was opened fully within 5 seconds, provided the oil pressure was in excess of 100 lb. per sq. in. D.T.D. 230 fuel was used. The examination of the engine after this severe test showed it to be in first-class condition (see Figs. 1–3). There were no stuck piston rings, wear was negligible, and deposits in crankcase, crankshaft, oil holes, etc., were slight. In comparison with a motor oil formerly tested, an engine of the same size but developing less than half the horse-power, was dirty, sticky sludge deposits showing inside the pistons, crankshaft, crankcase, etc., while a number of oil drain holes in pistons and rings were partially choked. Two piston rings were stuck. No signs of lack of lubrication of any bearing surfaces could be observed. The comparison of the above oil tests may be regarded as extreme, but they represent results obtained at an interval of six years.

Although great improvements have been made in lubricants, credit for the remarkably improved condition of modern engines on stripping after severe bench tests must be given to engine builders because of the vast amount of research they have conducted on the improved cooling of cylinders, better piston and ring designs, more accurate manufacture, better finish of engine parts, and stiffer designs of connecting rods, crankshafts, bearing mountings, etc. Internal deposits in engines are much influenced by piston ring leakage, and the elimination of excessive cylinder, piston, and ring distortion, due to improved design and manufacture, has assisted markedly in securing present-day results. The use of nitrided cylinder barrels has also shown beneficial results.

Deposits and stuck piston rings in engines are often assumed to be the direct result of oxidation of the lubricant; it is, therefore, of interest to note that Auld (1936) points out that crankcase sludges are largely

composed of adventitious matter, metallic oxides, carbon, etc., which are only partly and indirectly oxidation products. Stuck piston rings result in high piston temperatures which have well-known effects on engine condition. He suggests that heat, even possibly more than oxidation, is responsible for the decomposition of lubricating oils into resinous materials which gradually form in piston ring grooves.

Doubts have been expressed whether some of the more modern aircraft engine oils which have been refined to give high stability at elevated temperatures provide the necessary lubricating properties. Severe bench tests showing low rates of engine wear adequately dispose of such fears. One such oil compared with an oil to average Air Ministry specification was tested in the National Physical Laboratory machine by the Jakeman method (King 1934). Tests on plain bronze bearings 2 inches in diameter and 2¼ inches long, with a bearing pressure of 1,000 lb. per sq. in., and a speed of 1,300 r.p.m., the shaft being of steel, gave the following results:—

	Seizing temperature, deg. C.			Maximum temperature, deg. C.
Aero-engine oil . .	270	280	326	326
Average D.T.D. 109 oil .	235	260	255	265
Running period, hours .	10	23½	50	—

These results show that high lubricating properties need not be sacrificed in an aviation lubricant which has high resistance to oxidation, and may in fact be decidedly superior to an ordinary oil. Test bench trials of severe character carried out in engines of varying types, have shown the superiority of the former oil.

Interest has been shown in increasing the severity of oxidation tests (Air Ministry method) by raising the temperature of oxidation above the normally accepted figure of 200 deg. C. Auld (1936) quotes some interesting figures in this connexion, as follows:—

Oil	Carbon residue (Ramsbottom) with rise of temperature			
	Before oxidation	Temperature of oxidation, deg. C.		
		200	230	250
Modern aviation oil,* per cent.	0·22	0·68	1·99	2·98
Ordinary type of D.T.D. 109, per cent	0·47	1·4	3·32	5·58

* 100 per cent mineral oil.

It is well known that some oils show great increases in viscosity under severe oxidizing conditions, and the following comparisons are quoted from the same source:—

Oil	Increased viscosity at 100 deg. F. with rise of temperature				
	Temperature of oxidation, deg. C.				
	200	210	220	230	240
Modern aviation oil . .	1·38	—	—	—	3·84
Ordinary type, D.T.D. 109 oil	1·6	2·04	3·15	5·37	17·0

The stability of a recent aircraft engine oil under severe bench test conditions is shown hereunder:—

Tests	Unused oil	After 100 hours' bench test
Specific gravity at 15 deg. C. . . .	0·882	0·885
Organic acidity, gm. KOH per 100 gm. oil	0·002	0·004
Coke value, per cent. . . .	0·18	0·55
Coke value after blowing . . .	0·63	—
Oxidation (viscosity ratio) . . .	1·38	—
Ash, per cent	Negligible	0·09

Redwood viscosities of samples taken every 25 hours from a Bristol "Pegasus III M.3" engine during a 100-hour test are as follows:—

Test	Unused oil	Used oil after			
		25 hours	50 hours	75 hours	100 hours
Redwood viscosity at 70 deg. F., seconds	3,245	3,087	3,187	3,206	3,274
Redwood viscosity at 200 deg. F., seconds	100	99	100·5	100	102
Suspended carbon, per cent	Nil	0·16	0·31	0·27	0·31

This engine was developing 800 b.h.p. at 2,350 r.p.m., using 87-octane (D.T.D.230) fuel, the oil inlet temperature being 70 deg. C.

Deposits in engines have become more pronounced under extreme works test conditions with the wider adoption of leaded fuels. Samples of deposits taken from a modern engine tested with the propeller at ground level show the following analysis:—

Engine part	Piston crown	Valve necks	Crankpin
Carbon, per cent	35·1	75·9	14·7
Lead oxide, per cent	55·8	11·8	47·1
Silica, per cent	1·0	0·65	1·2

A carbon deposit from piston crowns after test, using non-leaded fuel, contained the following percentage amounts: carbon 74, oil 22, iron oxide 1·7. Analyses of crankshaft deposit show considerable variation in lead and carbon content, according to the type of unit and conditions of test. The contents of iron, copper, and other metallic oxides also show considerable variation. Leaded fuels are essential to high engine performance, and the lead oxide deposits in engines are a natural result.

Wilkinson (1933–4) outlined as early as 1934 the advantages of small-capacity cylinders operating at a high number of cycles per minute. The authors have seen some very interesting (Napier 24-cylinder "Dagger") engines stripped after test under high-power output conditions (Figs. 4–6). The internal cleanliness of these engines was possibly due to the relatively large cooling surfaces and the use of an oil of high heat stability. The oil temperature rise (in and out) is comparatively moderate—about 20 deg. C. under test bench conditions—while the rate of oil circulation is high.

It is interesting to note that rates of oil circulation in high-powered engines vary considerably according to type, some "line" engines having oil circulations up to four times faster than the radial types, due to the larger number of bearing surfaces to be lubricated and cooled.

Future engine developments point to still higher horse-powers and speeds from power units of existing dimensions. The Rolls-Royce "Merlin" has an official rating of 1,050 b.h.p., while the recently announced Bristol "Hercules" Sleeve Valve (14-cylinder 2-bank) gives a maximum of 1,325–1,375 b.h.p. The 24-cylinder Napier-Halford "Dagger" is claimed to give 49·7 h.p. per litre of cylinder capacity. Weights (dry) of existing high-performance engines seem to lie

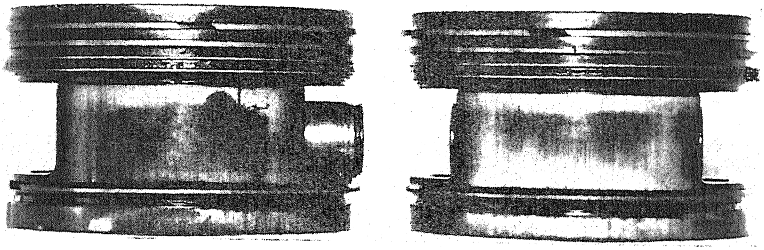

Fig. 1. Typical Piston of the Bristol Pegasus III, M.3 Engine (Supercharged), showing Non-thrust and Thrust Sides, after 100 Hours' Bench Test

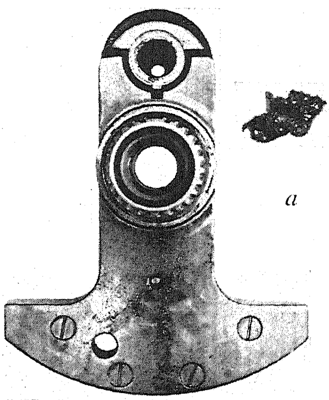

Fig. 2. Crankshaft (from the Same Engine as Fig. 1) showing the Small Sludge Deposit (*a*)

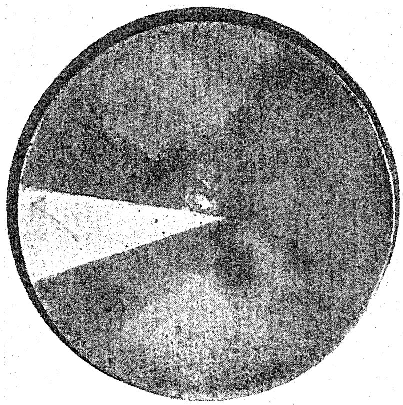

Fig. 3. Piston Crown (from the Same Engine) with Carbon Partly Removed

Fig. 4. Piston from the Napier "Dagger" Engine (Supercharged), after 60 Hours' Bench Test

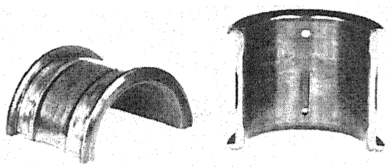

Fig. 5. Lead Bronze Big-end Bearing (from the Same Engine as Fig. 4) after Test

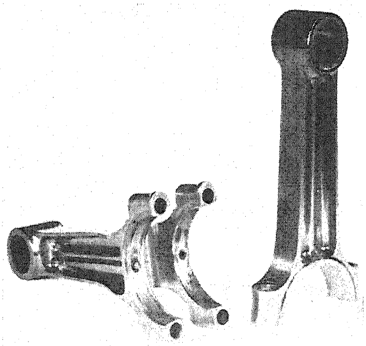

Fig. 6. Connecting Rod (from the Same Engine as Fig. 4) after Test

between 1·38 and 1·16 lb. per h.p for units of different types between 800 and over 1,000 h.p.

The method of installing engines in aeroplanes designed for speeds exceeding 250 m.p.h. seems likely to lead to higher outgoing oil temperatures, owing to improved streamline design of engine cowlings. Speeds in flight tend to increase substantially and thus high outgoing oil temperatures enable oil cooling to be efficiently controlled without recourse to coolers giving excessive head resistance, as demonstrated by the Bristol "Blenheim" with air-cooled engines and the Fairey "Battle" fitted with a liquid-cooled Rolls-Royce "Merlin" engine.

New fuels of the 100-octane type are designed to provide considerably greater power from power units of existing cubic capacities; and lubricating problems are expected to increase in severity, which in turn may necessitate further developments in aero-engine lubricants.

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References

AULD, S. J. M. 1936 Jl. Inst. Petroleum Technologists, vol. 22, p. 62; vol. 22, pp. 72–73.

Bristol Review 1935, Engine Issue, July, No. G, p. 6.

FEDDEN, A. H. R. 1934 Jl. Inst. Petroleum Technologists vol. 20, p. 100.

FOORD, F. A. 1929 Jl. Roy. Aeronautical Soc., vol. 33, p. 1089, "Lubrication of Aircraft Engines".

KING, R. O. 1934 Jl. Inst. Petroleum Technologists, vol. 20, p. 100. "Oxidation, Lubrication and the Blending of Mineral Oils to obtain Maximum Lubricating Value".

WOOD, H. 1936 Trans. Soc. Automotive Eng., vol. 39, p. 267.

WILKINSON, G. S. 1933–4 Proc. Inst. Automobile Eng., vol. 28, p. 311.

THE NATURAL AND ARTIFICIAL AGEING OF AUTOMOBILE ENGINE OILS

By Dr.-Ing. habil. E. H. Kadmer.*

The tests made covered (*a*) the ordinary use of various lubricants in different vehicles running on different fuels, and (*b*) artificial ageing of heated oil by the action of air and oxygen.

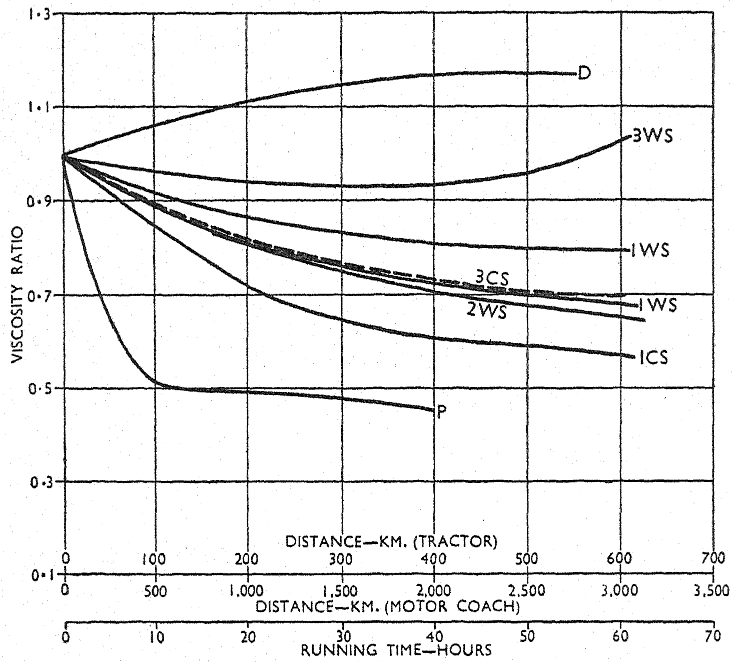

Fig. 1. Data from Fifty Driving Tests

1 Petrol. 2 Petrol+alcohol. 3 Petrol+benzene+alcohol.
P Petroleum. WS Warm season. CS Cold season.

Fig. 1 summarizes graphically the data obtained from 50 road tests. As oils of very different viscosity were used, the change in the viscosity could only be expressed as a ratio plotted against distance or hours of running. The curves are valid as a survey, but do not

* Munich.

clearly define suitability. The general inference is that with ordinary petrol in summer, the viscosity of the lubricant hardly decreases more than 20 per cent, but in winter it falls by 40–50 per cent during one filling of oil. With a petrol-alcohol mixture, under similar test conditions the oil dilution seems to be somewhat greater, the reason being that the added alcohol leaves the engine colder, thus favouring dew formation and consequently oil dilution. With the triple mixture the engine is always hot and oil dilution is least evident, even in winter. The fall in the viscosity of the lubricant with carburettor service in land and forest tractors burning heavy fuels (kerosene) is very severe and eventually leads to mechanical trouble. Many tests have shown the important fact that the fuel content of used oil is not derived from dew precipitation of already vaporized fuel, but only from particles existing solely as a mist of fine drops dispersed in air.

Oil dilution therefore depends on the dispersion of the fuel-air mixture, and it will be less the better the fuel is stably dispersed in the intake air. Obviously no rigid mathematical relations can be established between oil dilution, flash point reduction, and the fuel content of the used oil. To determine the fuel content in the oil is itself difficult, as there is no definite dividing line between the " boiling tail " of the fuel and the first fraction of the oil, particularly in lubricating oil diluted with kerosene. The flash point of a used motor oil depends not only on the degree of dilution but even more on the nature of the diluent. Small quantities of low-boiling constituents in the oil lower the flash point more than larger amounts of high-boiling fuel residues.

In Diesel service the change in the viscosity of the oil is different, often seeming at first sight to be " oil thickening " (Fig. 1, curve D). However, as much soot is produced in Diesel service, it can be assumed that this oil thickening is really due to sooting, as occurs in vehicles using wood or charcoal gas according to the degree to which the gas is cleaned, and large amounts of unburnt or unburnable substances get into the crankcase oil.

When acid materials appear in used oils it is usually assumed that they have been formed from the oil itself, but it is also possible that they originate from incomplete combustion of the fuel. The latter possibility gains in probability when it is considered that, particularly with a cold engine, incomplete combustion of the fuel gives rise to acid compounds. There is a certain relation between oil acidification and asphalt formation, since the acidification first increases, and then the asphalt content, partly at the expense of the acidification. This raises the question how far asphaltic substances are formed by

acidic constituents of lubricating oil. Here it should be remembered that a fair proportion of the acid substances adheres particularly to the dark suspension in used motor oils. The combustion of alcoholic motor spirit leads to acidity in the lubricant, while wood gas, from which acid tar gets into the oil, also causes acidification of the oil.

In dealing with sludge formation a distinction must be made between combustible and incombustible solid foreign bodies in used motor oils. The data are so abundant that a useful and conclusive survey seems almost impossible at first sight, but this does not apply if two fundamental considerations are kept in mind:—

(*a*) The combustible foreign bodies in used motor oils are formed by soot, oil carbon, and asphalt; they may originate from the oil itself, but must in the main be traced back to the always incomplete combustion of the fuel. Now the correct relation between test results on different aged oils is not revealed at all by reducing the values to a uniform driving distance. As a basis of comparison the author has used the ratio of fuel consumption to oil consumption, because careful observation has shown that vehicles carrying a large charge of oil are less subject to oil contamination than those with smaller charges. Apparently the speed of oil circulation is not so important, as oil deterioration is due to a much smaller extent to use of the oil, for which the speed of circulation, mode of cooling, etc., would be of importance.

(*b*) The non-combustible foreign bodies in used motor oils consist of dust and metallic particles. To find a basis for comparison the author proposed, many years ago, the relation with the linear travel of the piston for all cylinders. P. Schneider (Berlin) has improved on this in recommending the area swept by the piston as a basis of comparison. This cylinder area, in square kilometres per 100 kilometres driving distance is given by:—

$$F=\frac{2h.d.\pi.n.z.60.100}{10^{10}.g}=\frac{12h.d.\pi.n.z}{10^{7}.g}$$

where h is the stroke in centimetres, d the cylinder bore in centimetres, n the revolutions per minute, z the number of cylinders, and g the driving speed in kilometres per hour.

Further, if one calculates from 100/F the driving distance of the vehicle which corresponds on the average to a piston sweep of 1 sq. km., then the abrasion values acquire correct importance through the determination of the ash content. The bearing wear has not been considered, of course.

Regarding oil contamination by soot from the fuel, it can be proved

that it is directly related to incomplete fuel combustion. The degree of completeness of fuel combustion depends undoubtedly upon the C : H ratio of the fuel, and so we find lowest oil contamination through soot with a high C : H ratio (methyl alcohol, liquid gas, engine methane, fuel oils, light motor spirits) and high contamination with a low C : H ratio (benzene, Diesel oils, generator gas). Further all the observations suggest that the asphalt content of used oils is primarily due to incomplete fuel combustion; this also depends upon the C : H ratio of the fuel. Since with clean combustion (C : H above 1 : 2) the amount of soot and asphalt in a used oil is always small, it is evident that the development of such substances from the oil itself is to be considered as small, and that the wear of the oil itself by no means plays the important part in practical running that has hitherto been assumed.

As regards lubricating power of the oil and wear, there is nothing to prove that oils which popularly are ranked high show a low content of metallic particles or asphalt. The ash content in oil for 1 sq. km. piston sweep in carburettor service averages between 0·05–0·10 per cent, and is higher only in aero-engines with the parts under greater stress. Further, in Diesel service, the average values are 0·08–0·15 per cent, and higher values occur only when more mineral dust is introduced with the intake air. Also, it is incorrect to believe that high cylinder wear necessarily implies high ash content of the oil, for only the finest abraded particles remain in suspension in the oil, coarser particles being deposited and thus do not appear in the laboratory examination at all. Thus experience teaches that run-out bearings do not increase the ash content of the oil.

In artificial ageing, measured amounts of air or oxygen are passed through the heated oil, in some cases in the presence of catalytic accelerators. During such ageing there is always a thickening of the oil, which increases with increasing duration of the test, the development of acidity, and a more or less pronounced increase in the asphalt and oil carbon, substances derived solely from the oil. Oils differ considerably in oxygen stability, but this is not necessarily equivalent to quality. There is a great difference between oils subjected to the influence of fuel residues and oils artificially aged. Used oils, even with the greatest contamination, can be regenerated with sulphuric acid and fuller's earth or selective solvents and made equal in value to the original oil. Oils aged artificially, however, are by no means capable of being regenerated. For this reason, artificial ageing cannot be used alone to judge the quality of a motor engine oil. It may be different with compressor oils, although the test arrangement does not even here come near to normal conditions of use. In this

TABLE 1

Use		Sp. gr. at 20 deg.C.	Viscosity in Engler degrees at			Viscosity pole height	Viscosity index	Flash point, deg. C.	Sapon. value	Asphalt, per cent	Carbon, per cent	Ash, per cent	Colour (Ostwald scale)
			20 deg. C.	50 deg. C.	100 deg. C.								
I	*f*	0·910	109	13·8	2·27	2·5	—	—	0·17				7·6
	a	0·912	104	13·7	2·25	—	—	—	6·38	0·076	0·210	0·030	10
	r	0·907	99	13·2	2·25	2·5	—	—	3·42				6·8
II	*f*	0·882	113	17·0	2·75	1·9	—	—	0·71				7·0
	a	0·898	80	13·4	2·45	1·9	—	—	2·55	0·202	1·741	0·102	10
	d	0·893	107	15·3	2·50	2·1	—	—	0·66				8·4
	r	0·884	112	16·8	2·70	1·9	—	—	1·16	0·028	0	0	5·4
III	*f*	0·848	38	6·5	1·72	2·2	+57	217					4·0
	a	0·860	27	5·4	1·69	2·0	+82	100	5·06	0·052	0·286	0·127	10
	r	0·855	29	5·7	1·71	2·0	+82	141	1·39	0·008	0	0·002	5·5
	ss	0·877	10	2·7	1·37	2·0							6·5
IV	*f*	0·878	124	17·6	2·75	2·0	+93	263	0·07				
	a_1	0·869	56	9·3	2·02	2·1		157	4·55	0·212	0·614	0·158	
	r_1	0·883	107	15·4	2·51	2·1	+83	235	1·17				
	$r_1—a_2$	0·883	55	10	2·12	1·8	+94	100	6·38	0·120	0·905	0·102	
	r_2	0·883	118	17·4	2·74	2·0	+93	242	1·32	0	0	0	

connexion the importance of coking tests (Conradson or Ramsbottom carbon test) may be diminished.

That lubricating oils are not internally stressed to the extent that has hitherto been assumed is clearly shown by the results of refining used motor oils. It is not intended to discuss oil regeneration here, but as all the conclusions from the study of natural and artificial ageing of motor engine oils clearly indicate that it is at any rate technically possible to regenerate such oils so perfectly that they are practically equal in value to fresh oils, the point is illustrated by the figures given in Table 1.

Table 1 refers to the following tests:—

I. Small plant, only with bleaching filtration under pressure; *f* fresh oil; *a* aged oil after 100 hours in Diesel engine; *r* regenerated oil.

II. Small plant, with sulphuric acid refining and distillation of fuel residues; *a* aged oil after 40 hours in aero engine; *d* distilled oil, partly bleached.

III. Laboratory test: *a* aged oil after 1,750 kilometres in motor coach; *r* regenerated oil, acid treatment, and bleaching; *ss* solvent extract from regenerated oil.

IV. Industrial lubricating oil regeneration: a_1 aged oil, after 30 flying hours; a_2 second used oil after 4,000 kilometres in motor coach; r_1, r_2 regenerated oils.

APPLICATION OF MAGNETIC FILTERS TO LUBRICATING SYSTEMS

By L. H. de Langen*

In establishing the hydrodynamic theory of lubrication it was assumed (1) that the surfaces sliding on each other are perfectly flat, and (2) that the lubricating material is a pure liquid. In practice neither of these two conditions is satisfied entirely. Later investigations have shown that most surfaces obtained by ordinary means, even when viewed with moderate enlargement, resemble a mountain landscape. It is essential that the heights of the mountains in this landscape should be roughly of the same order of magnitude as the thickness of the film of lubricating oil. It is not surprising that, when these surfaces are caused to slide over each other, very little remains of the theory of lubrication; the layer of oil is broken up and considerable wear is the result. The author has often noticed that even in very accurately turned bushes, after an axle had run in them for a number of hours, the diameter became a few hundredths of a millimetre larger; the peaks of the hills had been worn down. Through this wear, the second assumption, namely, that lubrication must be effected by means of a pure liquid, collapses. The particles of metal liberated through wear circulate with the liquid and cause fresh wear. This experience leads to the following paradox: a bearing must be lubricated with as little oil as possible. If through wear the oil has become impure, it is better to use little rather than much oil, in order that but few solid particles can take part in the further wear of the surfaces. In a certain machine with two approximately equally loaded bearings, one of which received very much oil and the other through chance circumstances practically none, the former showed by far the most wear.

Therefore, to improve lubrication and reduce wear, firstly, the aim must be to obtain more ideal surfaces and, secondly, means must be sought for removing the metal particles suspended in the oil before they can cause further wear. By using modern methods of treatment such as lapping and honing, it is possible to give the surfaces of axles and bearings such a finish in the ordinary course of manufacture as to meet the first requirement.

The object of this paper is to describe a method developed by the N. V. Philips' Gloeilampenfabrieken for continuously clearing the oil of metal particles, thus fulfilling the second requirement. The

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principle of this method is that during circulation the lubricating oil is continuously passed through a powerful magnetic field, so that the iron particles are attracted by one of the poles of the magnet and extracted from the oil.

Modern magnet steels are such that permanent magnets of not too great dimensions can give fields so strong that even very small iron particles can be extracted by them. Fundamentally, a magnetic oil filter consists of a cylindrical permanent magnet, placed in a cylindrical space surrounded by iron walls (Fig. 1). The permanent magnet A is

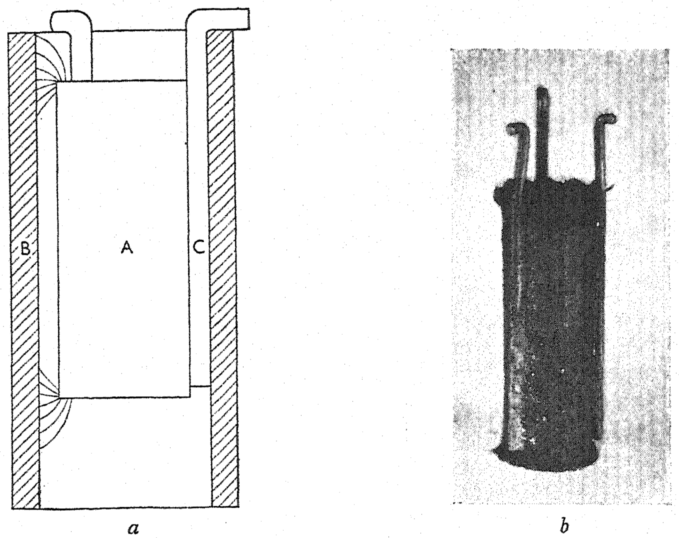

Fig. 1. (*a*) A Simple Magnetic Oil Filter. (*b*) Small Magnet with Particles of Iron removed from Oil

placed in a steel pipe B. The brass wires C soldered to the magnet hold it in the middle of the pipe. If oil containing iron particles flows through this pipe, then the iron is extracted, particularly at the upper side of the magnet, at the places of greatest concentration of the lines of force, as can be seen in Fig. 1*b*, which shows a soiled magnet. An oil filter of this design is very satisfactory, but rather limited in capacity; owing to the dirt settling in a thin ring, the filter is soon full.

In an improved type of oil filter (Fig. 2) the permanent magnet A is attached to a cap B screwed into the iron casing C. A brass tube D with holes E is placed round the magnet to guide the oil, which enters at F and leaves again at G. This construction has three important

advantages. In the first place a good magnetic circuit is formed with merely one air gap, so that for a given length of the permanent magnet the magnetic field is as powerful as possible. Second, the spherical pole of the magnet is placed in such a way with respect to the casing that the iron particles deposit themselves on the whole spherical surface and consequently the filter can collect much iron dust. Third, the shape of the air gap is such that the inflowing oil flows in roughly the same direction as that in which the iron particles are attracted. The

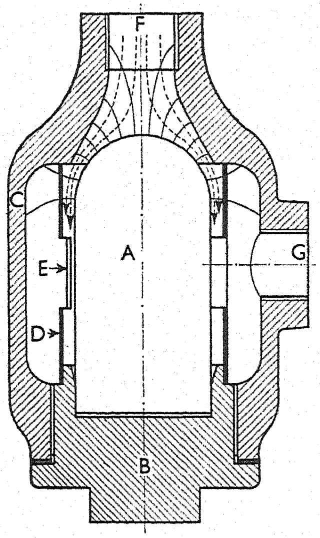

Fig. 2. Section of Oil Filter

——— Magnetic lines of force.
- - - - Current lines.

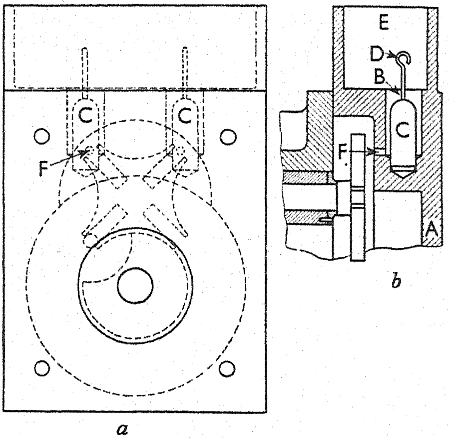

Fig. 3. Maltese Cross Incorporating Magnetic Oil Filters

In (*a*) the magnets are drawn in dotted lines; (*b*) shows a section of the magnet.

lines of force and the current lines cut each other at small angles; in this way the iron particles are subjected over a long path to the attraction of the magnet.

Several magnetic filters have for some time been in use at the N. V. Philips' Gloeilampenfabrieken. The first general application was found in the film projectors of the author's firm which are now equipped with magnetic oil filters. In two ways a successful effort has been made to reduce to a minimum the wear of film projectors. The first step was to give the parts smoother and more accurate surfaces.

Much has been achieved in this direction. All shafts and bearing bushes are lapped or honed, and also the Maltese cross is finished to a greater smoothness and higher precision (tolerance 1 and 2 microns) by means of a special lapping treatment. The gearwheels, however, are still subjected to ordinary treatment, which is effected by hobbing with a small feed. Lapping or grinding of these small gearwheels is not yet possible. When running-in the gearwheels, wear was produced that acted as grinding material and spoilt the fine surfaces of the lapped shafts, the Maltese cross, etc. Here the use of simple magnetic filters brought about a great improvement. In the casing of the Maltese cross two of these filters are built in. Fig. 3 is a cross-section of one of these filters. A is a part of the Maltese cross casing, B a hole in which the permanent magnet is placed, D a wire of non-magnetic material that has been cast in, E an oil reservoir that is continually

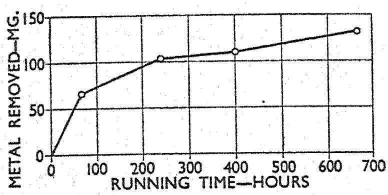

Fig. 4. Wear of a Projector Equipped with a Magnetic Oil Filter

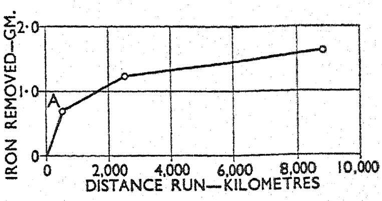

Fig. 5. Wear of an Automobile Engine

filled up again by the oil pump. The oil flows through the hole F in the casing of the Maltese cross. Of the oil sent through by the oil pump only that part intended for the Maltese cross is filtered. However, as the oil circulates rapidly the other oil soon reaches the Maltese cross. This method is very satisfactory; when a magnetic filter is placed in a projector whose oil has become dark in colour owing to particles of iron worn off, the oil again becomes light in colour after some time. The length of the iron particles trapped has been found to vary between 0·4 and 4 microns. After running in, wear quickly decreases, as will be seen from Fig. 4. The reduction of wear is of particular importance, for a Maltese cross that shows play owing to wear begins to hit harder and then wears out still more rapidly.

Interesting results were obtained with a magnetic filter in a motor car engine which had been overhauled, the cylinders having been honed and the pistons fitted with new piston rings. A filter of the type shown in Fig. 2 was inserted in the pressure oil pipe. After the car

had run 500 kilometres the filter (Fig. 6) already contained about 673 mg. of iron particles (Fig. 5A) varying in size between 0·4 and 6 microns. That wear was reduced, can be seen from Fig. 5. Here wear shows the same trend as with cinema projectors. For several industrial machines oil filters have likewise been used successfully. An oil filter fitted in the lubricating system of a steam turbine was the only one that was not very successful.

It is possible to follow several trends in the application of magnetic oil filters. Particularly in not very large machines, there is a practical advantage in incorporating an oil filter in the central pressure

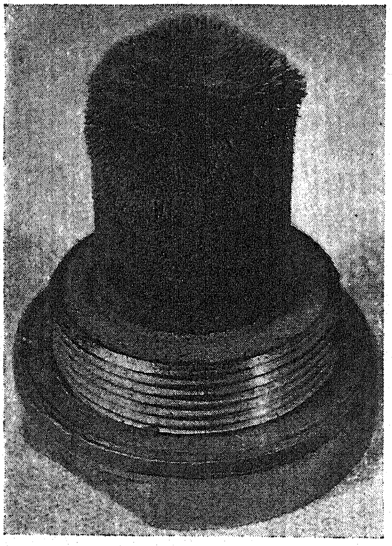

Fig. 6. Magnet with Particles of Iron

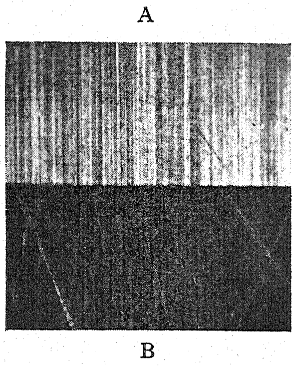

Fig. 7. Photomicrograph with Lateral Illumination of the Surface of a Shaft
×102 dia.

A. Unworn part. B.Worn part.

oil pipe. In larger machines with very large quantities of circulating oil it suffices to purify merely part of the oil by placing the filter in a branch of the pressure pipe. Another possibility is to fit a simple oil filter immediately before each lubricating point. In many cases it is not difficult to arrange the holes for lubrication in such a manner that a small magnet can be placed in them, in the manner shown in Fig. 3.

A magnetic oil filter can only arrest particles of iron; particles of bronze or whitemetal remain suspended in the oil. Fortunately the parts which produce the greatest initial wear, such as gearwheels, piston rings, cylinders, etc., are usually made of iron or steel. Cast

iron can often be employed to advantage where bronze is still often used. Thus, for instance, the use of cast-iron bearing bushes in cinema projectors was successful, whilst a further advantage was the possibility of catching any worn particles from the surface of the bush in a magnetic filter.

Wear owing to metal particles in lubricating oil has been further investigated with the following simple test. A silver-steel shaft of 10 mm. diameter, lapped in normal production, was rotated for $1\frac{1}{2}$ hours in smooth bronze bearing bushes. In order to ensure that lubrication would not be unfavourably influenced by an excessively small clearance, liberal play was allowed, namely, about 0·06 mm. (difference of the diameters). The shaft ran at about 1,000 r.p.m., being driven by a string belt; the tension of the string was the only load on the shaft. The bearings were lubricated with oil that had been made impure artificially with iron particles taken from the magnetic oil filter of a projector. At the end of the test the axle showed ring-shaped grooves that were visible to the naked eye. With a special Busch comparison microscope for the examination of surfaces a photograph (Fig. 7) was taken of the worn part of the axle and of the part that was outside the bearings. Section A (Fig. 7) shows the undamaged part of the shaft. Here the fine interlacing scratches of the polishing treatment can be seen clearly. It is not difficult to get an even finer surface that appears practically black in the microscope, but in ordinary practice this extreme quality is not necessary. Section B shows the worn shaft. Nothing can be seen of the original polishing scratches; in their place there are now much coarser grooves produced by wear. This simple test, which can easily be repeated, proves in a convincing manner that the purification of circulating oil used for lubrication deserves every consideration.

LUBRICATION SYSTEMS FOR DIESEL ENGINES

By Professor A. L. Mellanby, D.Sc., LL.D.*

The problems associated with the lubrication of Diesel engines may be considered under two headings: (1) the lubrication of the cylinder, and (2) the lubrication of the bearings. Each is of great importance, and upon its correct solution the life and efficiency of the engine largely depend. Even to-day cylinder and piston wear is in some cases greater than is desirable, and while bearing lubrication may be said to have become fairly standardized, yet the necessity for reducing weight and, consequently, bearing areas, will on occasion present the designer and operator with problems which can only be solved by a close study of fundamental principles.

When once an efficient system of lubrication has been evolved, failure may occur from the use of an unsuitable oil, and in the early days of the marine Diesel engine much trouble resulted from this cause. But service experience combined with research on the part of the oil blenders has greatly reduced any difficulties that might thus arise, and grades of lubricant carefully selected to stand up to the most trying conditions are now readily available.

Cylinder Lubrication. For cylinder lubrication, the ideal would be to obtain on the inside walls an unbroken film capable of acting both as a lubricant for the piston rings and as a seal to the passage of the products of combustion. When it is remembered that the gas temperatures during the ignition period may be in the neighbourhood of 1,500 deg. C., the difficulty of maintaining such a film will be readily recognized. Part of the oil is burnt away during each cycle, and this must be constantly renewed. Care must, however, be taken to ensure that only sufficient oil to maintain the film is injected, as, otherwise, other troubles will soon arise. Excess oil will be partially burnt, forming a sticky deposit on the piston rings, while this in turn will ultimately lead to the discharge of partially burnt lubricant into the crankcase, where it will mingle with the bearing lubricating oil.

In early examples of Diesel engines, oil was introduced into the cylinder by means of a small force pump, through a pipe encircling the cylinder near the bottom and provided with a number of branches each of which communicated with a hole drilled through the cylinder walls. At the present time the general practice is to connect each supply hole direct to its own force pump, although in some cases one

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pump is used to supply two holes. The number of holes employed and their position round the cylinder vary with different makers. In some cases the holes are spaced approximately equally round the circumference, while in others they are grouped near the ends of a diameter drawn from back to front. In the case of one well-known firm, the total number of oil holes is fixed by the cylinder sizes, so that for cylinder diameters between 300 and 500 mm. there are four

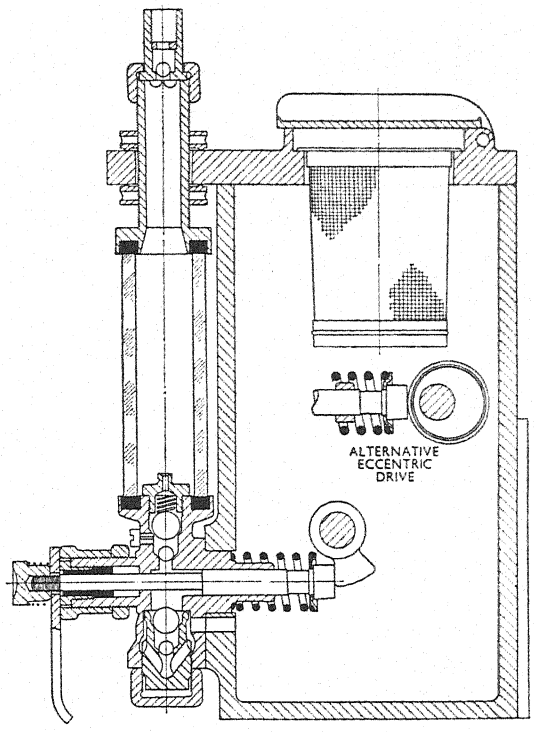

Fig. 1. Type of Lubricating Pump

holes, between 500 and 600 mm. six holes, and above 600 mm. eight holes. Half the holes are situated at the front side of the cylinder and half at the back.

It will be noted that, since the oil delivered per stroke of the pump will be very small, and since even this small quantity should be capable of adjustment, the conditions to be fulfilled by the lubricating pump will be somewhat exacting. The type of pump in most general use is shown in Fig. 1, from which it will be seen that it has a differential

plunger with a sight feed on the delivery side. With the differential plunger, it is possible to discharge very small amounts of oil per stroke with a plunger of relatively large diameter. The stroke and consequently the delivery is regulated by the adjustment shown, which brings the end of the plunger nearer to or away from the centre of the driving cam. Care must be taken that a reasonable relationship between plunger diameters and stroke is obtained, as otherwise the

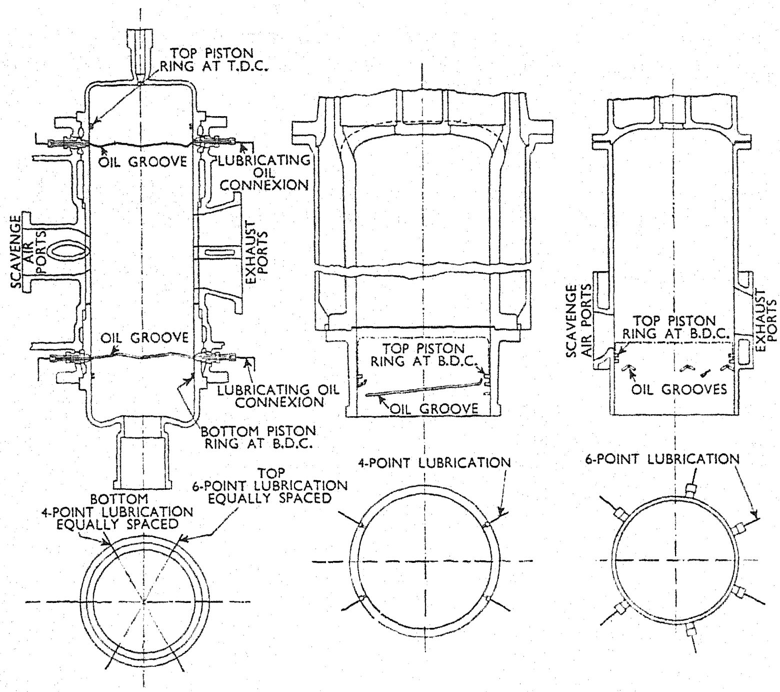

Fig. 2. Types of Cylinder Lubrication (Scott Engines)

pump stroke for the required oil delivery may be so small that close regulation is impossible.

At the delivery end of the oil supply pipe, some type of non-return valve is fitted. The non-return valve prevents oil from draining out of the supply pipe when the engine is at rest, and thus ensures an immediate supply to the cylinder when the engine is started. For complete lubrication of the walls, the oil should be evenly distributed by the piston. In the four-stroke cycle type the lubricating holes are near the bottom of the cylinder, and the aim of the designer is to time the period of delivery so that the oil injection takes place when the

holes have been passed by at least one piston ring on the downward stroke. In this way most of the oil is trapped between the piston and the walls and distributed over the whole length of the cylinder on the upward stroke.

With engines of the single-acting two-stroke cycle type there are appreciable differences between the practice of different makers. In some cases the oil injection holes are at the bottom of the cylinder below the exhaust ports. In other cases they are just above the exhaust ports, whilst the latest practice of Messrs. Sulzer Brothers is to place them at the top of the cylinder. In the Doxford opposed-

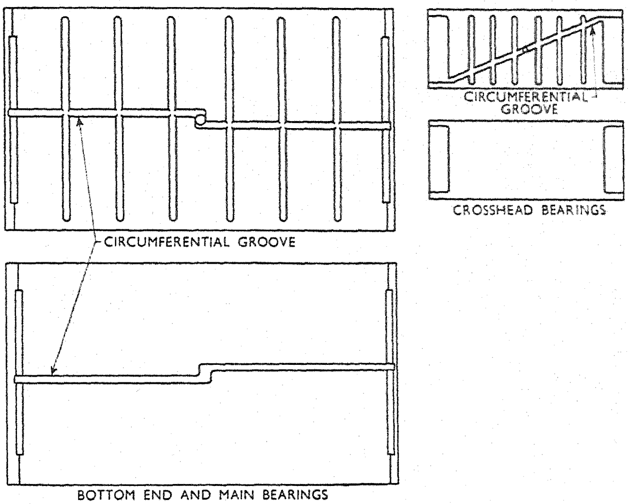

Fig. 3. Detail of Oil Grooves in Bearings (Doxford Engines)

Cross grooves are provided only on the loaded half of the bearings.

piston engine, it is arranged that injection of the lubricating oil takes place between No. 1 and No. 2 piston rings of each piston when they are at the end of the compression stroke. There are also differences in the type of grooves cut in the cylinder walls. Some examples are shown in Fig. 2. For double-acting two-stroke cycle engines, two rings of oil holes, one at the top and the other at the bottom, are employed.

Bearing Lubrication. With the forced system of bearing lubrication now universally employed, the problem is not so much one of getting the oil to the bearing but rather one of ensuring a proper circulation through the bearing, so that the heat generated by friction may be

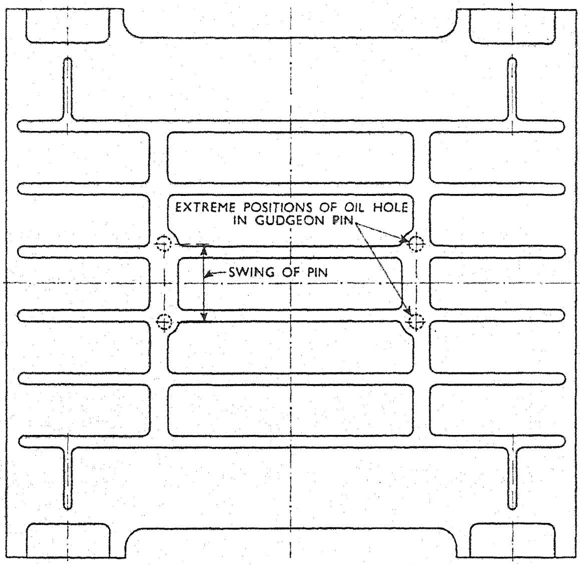

Fig. 4. Developed View of Whitemetal Surface (Scott Engine)

Oil grooves in bottom half of top half-bush of connecting rod.

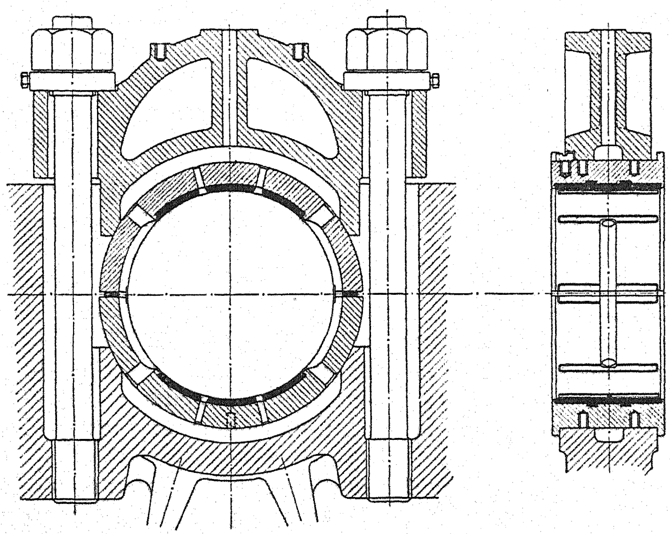

Fig. 5. Main Bearing (Burmeister and Wain Engine)

carried away by the oil. In early examples this latter point was not sufficiently recognized and, consequently, repeated changes were made in the number and positions of the bearing oil grooves until satisfactory running conditions were obtained. Practice is now more stabilized, but each manufacturer has his own ideas upon bearing lubrication, and some of the different methods of providing for the oil distribution and circulation will be seen in Figs. 3 and 4, which show the system adopted in the Doxford and Scott engines respectively. In the former it should

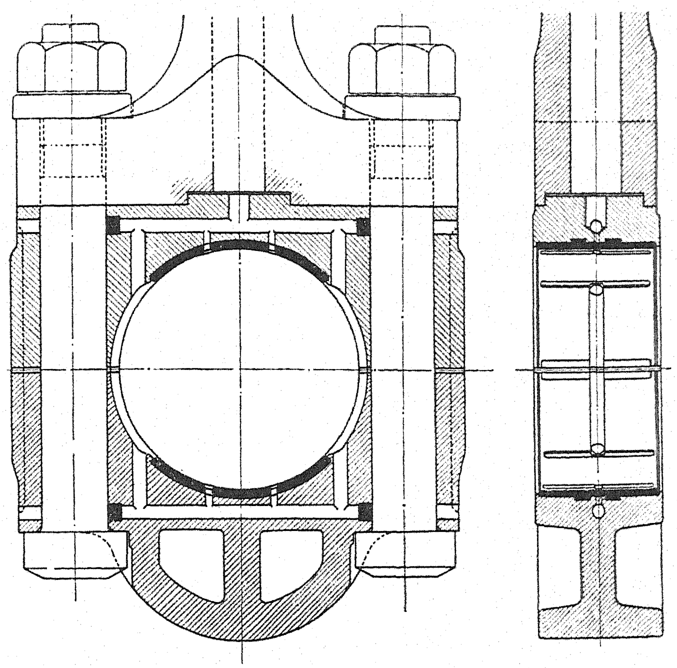

Fig. 6. Crankpin Brasses (Burmeister and Wain Engine)

be noted that the circumferential groove is stepped so as to avoid the formation of ridges on journals and crankpins. The same end is achieved in the top end bearing by making the circumferential groove diagonal and spacing the parallel cross grooves closer than the swing of the connecting rod.

In the usual lubricating system a main oil supply pipe running along the engine is provided with branches to each main bearing. From the bearing the oil passes through holes in the crankshaft and web to the crankpin, thence to the crankpin bearing and through a hole along the

centre of the connecting rod to the crosshead bearing, and in some cases to the crosshead shoes. The oil leaving the various bearings falls into the crank pit, and after being cooled and filtered is returned by the pump to the main supply pipe.

An example of effective bearing design for lubrication purposes is

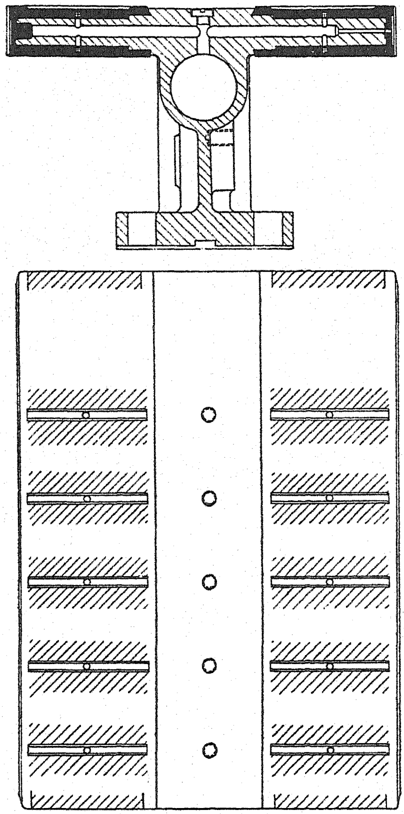

Fig. 7. Crosshead Guide (Burmeister and Wain Engine)

illustrated by Figs. 5 to 7, which show the system now employed by Messrs. Burmeister and Wain for their recent designs of two-stroke cycle engines. The main bearing is shown in Fig. 5, from which it will be seen that the oil supply is delivered to the bearing cap and a certain quantity is thus stored under pressure in the top and bottom pockets shown in the cover and bedplate respectively. Longitudinal grooves

scraped wedge-shape are cut in both the top and bottom halves of the bearing, and each groove is connected with a pocket under pressure by a hole, so that it is continually filled with oil. The grooves are continued quite close to the ends of the bearing. At the sides of the bearing, pockets are cut away in the whitemetal, and these communicate with the pressure spaces by comparatively large holes. The shaft has two radial holes, so spaced that at least one hole is always in communication with a pressure pocket during a revolution. Thus the central hole in the crankshaft is filled with lubrication oil under pressure, and this is led through holes in the web and the crankpin to the crankpin brasses. The design of the crankpin brasses is shown in Fig. 6, where it will be seen that in general principles the method of receiving and distributing the oil is similar to that employed in the main bearings. From the crankpin the oil is delivered through a hole along the connecting rod to the gudgeon pin brasses in which are cut grooves similar to those already described. In these engines a separate oil supply is delivered to the crosshead guides, shown in Fig. 7. The method of lubrication is shown clearly in the Figure and requires no further explanation.

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PRACTICAL LUBRICATION PROBLEMS FROM THE STANDPOINT OF THE MECHANICAL ENGINEER

By Eng. Lt.-Commander H. J. Nicholson *

Steam Cylinder Lubrication. It is true to say that more cylinder oil has been wasted than is effectively used for steam cylinder lubrication; often this is brought about by failure to consider or appreciate a few simple facts.

Ideal steam cylinder lubrication is effected by the spread of a uniform oil film over the contacting surfaces of valves and cylinders while injecting a minimum quantity of cylinder oil into the engine. To obtain satisfactory cylinder lubrication, certain factors must be considered:—

(1) Selection of correct type of oil.
(2) The possibility of impregnating the ingoing steam with atomized oil.
(3) The ability of the moving parts to spread the oil over the rubbing surfaces.

(1) If high-temperature steam is employed, then an oil capable of withstanding the prevailing temperature without decomposition must be selected. If low-temperature wet steam is used, an oil which will emulsify with the moisture is desirable. If the steam contains boiler impurities, this fact will have to be considered. When condensate is used for boiler feed or exhaust steam for process work, an oil capable of freeing itself readily from either must be selected.

(2) To impregnate the steam with oil, a suitable position in the steam pipes or passages must be selected. This point should be where maximum steam velocity is obtainable, avoiding as far as possible positions in the vicinity of sharp bends or intricate passages or where the steam changes direction. If high-temperature steam is employed, then the atomizer must be close up to the cylinder or valves in order to reduce the possibility of decomposing the oil before reaching the surfaces to be lubricated.

(3) Large "D" slide valves are often troublesome to lubricate because of the difficulty of introducing oil between the working faces. If, however, the oil can be satisfactorily introduced there is plenty of movement to spread it uniformly; oil-impregnated steam is usually most satisfactory. A Corliss valve, on the other hand, has very little

* The Vacuum Oil Company, Ltd.

movement, particularly if running with an early cut-off, therefore the moving parts themselves do not spread the oil so readily. This type of valve frequently presents lubrication problems, and it is not unusual to feed oil direct to the working faces in addition to impregnating the ingoing steam with oil.

Deposits in Steam Cylinders. Deposits in steam cylinders and valve chests frequently present a problem to operating engineers, and these may be brought about by a variety of causes. Usually the deposits are either directly or indirectly the result of overfeeding lubricating oil, and for this reason the most satisfactory method of feeding steam cylinder oil is by means of a reliable mechanical lubricator which is closely controlled, and can be relied upon to maintain an accurate delivery of oil over varying conditions, such as change of speed in the engine, variations in the viscosity of the oil due to temperature changes, and variations in the level of the oil in the lubricator container.

When overfeeding takes place in the presence of solid impurities brought over with the steam, there is always the likelihood that deposits may accumulate due to the solids mixing with the lubricating oil, and setting up oxidation of the latter due to the cooking action of the steam. This combination produces hard deposits containing carbonaceous material.

There is a variety of causes for these deposits. For instance, a change in boiler feed water can result in difficulties with internal lubrication. To cite an example: a 1,000 h.p. vertical triple-expansion steam engine operating at a steam pressure of 150 lb. per sq. in., and a total steam temperature of 450–500 deg. F., had operated quite satisfactorily for a number of years on a certain grade of oil, the consumption of cylinder oil being one pint in 19 hours. It was later decided to treat the boiler feed water with soda and lime. Previously, it had been the practice to open up the engine after each 1,000 hours' running. On opening up at the usual period, but after the feed-water treatment had been introduced, deposits were found in the high-pressure cylinder which had not previously occurred.

Mechanical conditions frequently give rise to the suspicion that lubrication is the cause of operating difficulties. For example, an investigation of excessive wear in the piston and liner of a large uniflow engine revealed the fact that no tail rod was fitted. The operators were persuaded to make this alteration despite the fact that they had spent considerable time and money in an endeavour to improve the lubricating arrangements. After fitting the tail rod to one engine, the trouble was immediately overcome, and four other engines were afterwards similarly equipped.

Engineers are frequently faced with the question: What is a suitable flash point of a steam cylinder oil, taking into consideration the operating steam temperatures? Mere consideration of the steam temperature in relation to the flash point of the oil can be misleading. First, it must be realized that the flash point test is purely a laboratory determination carried out under closely defined rules and under ordinary atmospheric conditions with sufficient oxygen to support combustion. In the steam cylinder, the oil is in the presence of an inert gas under pressure, and therefore the temperature at which combustible vapour will be given off is increased and, further, there is very little oxygen present to support combustion. Therefore, the oil can be used at much higher steam temperatures than the actual flash point of the oil with entirely satisfactory results.

Air Compressors. The types and sizes of air compressors vary over a very wide range, and one of the main problems in connexion with their lubrication seems to be the formation of deposits on the internal surfaces, usually accompanied by the gumming-up of piston rings and valves. The difficulty centres around two main factors: (*a*) the fitting of efficient air strainers to the air inlet, and keeping such strainers in a clean condition, and (*b*) control of the quantity of the correct type of oil fed to the internal surfaces, and the position at which this oil is introduced.

Whenever possible, the air inlet should be placed in a position where it will obtain clean, dry, and cool air. Also a definite routine for the cleaning of air strainers should be instituted. If strainers are neglected, they merely act as accumulators for impurities. If they are of the gauze pattern, the accumulation of dirt goes on until a fracture takes place, and the impurities are then permitted to go over to the compressor in bulk.

In the small type of compressor, cylinder lubrication is usually provided by oil thrown from the connecting rod bearings, and this is difficult to control. In the larger type, however, some means of introducing the oil to the internal surfaces is provided, usually consisting of some form of mechanical lubricator. It is essential that such lubricators should be capable of being adjusted to deliver closely regulated quantities of oil, because satisfactory lubrication of an air compressor cylinder is only achieved by injecting the minimum quantity of oil consistent with adequate lubrication.

In some small air compressors, arrangements are made for injecting the oil into the air stream at the intake, but for large units this method is usually unsatisfactory, as the oil is cold and air is a bad carrier of oil; therefore atomization of the oil and subsequent impregnation of the

ingoing air is not always achieved. In the larger units it is invariably found more satisfactory to inject the oil direct to the cylinder, at one or more points, depending on the size. The piston will then assist in spreading the oil. In vertical machines, the oil is sometimes fed to the nut dome in the cylinder covers. This method is not conducive to good spreading of the oil.

The lubrication of an air compressor cylinder differs from that of a steam cylinder or an internal combustion engine. The compressor oil remains in the cylinder over a very much longer period, because it is not, as in a steam engine, washed away by the moisture or, as in the internal combustion engine, consumed under the prevailing temperatures. Again, it must be realized that excessive quantities of oil injected remain exposed to the oxidizing effects of the heated air under compression. These conditions, combined with impurities brought in from the atmosphere, considerably increase the liability to form deposits when compared with any other type of unit. Therefore, in order to reduce deposits in air compressors to a minimum, it is essential to keep the quantity of oil used under close control, and provide the cleanest possible air supply.

A problem which sometimes faces the engineer is the risk of explosions in the discharge pipes and receivers of air compressors. In an attempt to minimize this danger, an oil of high flash point is sometimes demanded; often in such cases, an oil having an open flash point in the order of 500 deg. F. is specified. If there are no deposits then there is practically no risk of fires or explosions, because the trouble is usually started by such deposits becoming incandescent, and they invariably accumulate round discharge valves or pockets and in discharge pipes.

Oils having high flash points come into the steam cylinder oil class. They are sluggish and viscous and, therefore, more readily form pasty deposits when they come in contact with the impurities drawn in from the atmosphere. Oils of low viscosity, with consequently lower flash points, tend towards cleaner internal conditions; impurities in the incoming air are more likely to pass through the compressor with the lighter oil, and thus reduce the risk of forming carbonaceous deposits which may become incandescent.

Gears. The lubrication of enclosed gear units often presents a difficulty of an essential practical nature. These units perform a wide variety of duties, yet in many cases the maintenance engineer is left to guess what oil is required for their efficient lubrication, and can then only describe his wants as light, medium, or heavy gear oil. It will be agreed that the fundamentals of tooth pressure and pitch line speed,

consideration of which should alone determine the lubricant required, are known only to the designer, and it therefore seems advisable that gear makers should always attach a plate to every unit they manufacture, indicating the correct lubricant. This information they can readily acquire on their own test beds, and once the best relationship between tooth pressure, pitch line speed, and lubricant had been established the recommendation could be made for all other sets without further trial. A standard gear unit is generally suitable for a wide range of speeds with a correspondingly wide range of tooth pressures, and it is reasonable to expect that different lubricants will be required for different conditions. It is not possible, therefore, to make the selection on dimensions only, and the need for attaching a plate is further emphasized. Motorized gear units, and machines individually driven through gearboxes, either fixed or variable, are very prominent examples where the user generally requires definite guidance as to the correct type of oil for their lubrication.

General Machinery. In spite of the many modern improvements in the design of machinery the older methods of lubrication, that is the oil can, wick oilers, and drop oilers, are still much in evidence and, it might even be said, indispensable. It is accepted that in most cases, unless adjusted extravagantly, they give only that "half-degree" of lubrication known as greasy or boundary lubrication, whereas the hand oil can, being entirely at the disposal of an operator, is often neglected or at best used erratically. Consequently, it is very satisfactory to be able to report that observation, of necessity over a long period, confirms the laboratory discovery that the addition of a small percentage of free fatty acid confers on the oil the ability to remain much longer and more intimately in contact with relatively moving surfaces, with the result that bush bearings and such parts where this method of lubrication is usually provided have had a much longer life when these compounded oils were used in place of the previous straight mineral oils. In addition, tests have shown that where many machines on continuous production are installed, and are entirely hand-lubricated, for example, the looms in a weaving shed, these oils have required less frequent application, wear has been reduced, and—final criterion—the units of electricity consumed over a reliably long period were reduced by a small but definite percentage. The conclusion is that these compounded oils are most satisfactory for all erratic or partial lubrication, and they supply a solution to this old practical problem.

Lubrication of Heated Bearings. Several industries, notably the textile and paper industries, have calendering processes in which the

rollers or bowls are heated. Steam and heated oil, at from 300 to 500 deg. F., are commonly employed to circulate round the bowls, and the only possible entrance and exit for the heating medium is through the bowl necks or journals. As a result the lubrication of the journal bearings requires special consideration, and is best accomplished by installing a circulating oil system with an external cooler. Difficulties have been encountered, however, due to unsatisfactory bearing design. A bearing of this type must be contained in a housing and must have a well because the supply of oil, although not under pressure, is copious, since it is desirable to keep the temperature of the mass of the oil well below the temperature of the necks. The design of these oil spaces calls for careful consideration, and the aim should be to provide smooth flow from all parts to the outlet, or, in other words, to avoid all stagnant or partially stagnant regions. Bearings in which this point has not been carefully observed have been troublesome because deposits form and accumulate, necessitating expensive dismantling for cleaning and loss of productive time. The correctly designed bearing with its good oil flow allows any deposit which may be formed, to be collected by the flowing oil and removed from the system by a suitably placed strainer. Another important point with such bearings is that the circulating system must be driven independently of the machine, so that circulation can be established before the machine is started up and maintained after it is shut down. This is done so that lubricant is available immediately the neck begins to rotate, when probably the most wear takes place, and also to ensure that no oil will be left stagnant in the bearing as it cools down from working temperature, to form products of decomposition of the oil. The formation of carbonaceous deposits in these bearings under the severest working conditions is hardly avoidable, and whilst the above observances will do much to prevent difficulties, the oil selected for the circulating system should be specified for that duty, so that the supplier can provide an oil with minimum tendency to form deposits under high-temperature working conditions.

Diesel Engines. Experience with these engines has so increased in the last decade that it is no longer the rule to lay the blame for all the usual running difficulties on the lubricating oil. Engine design, improvement in materials, and more exact knowledge of the combustion process have all contributed so much to the present successful standards that the modern engine is not nearly so sensitive to the quality and type of lubricating oil as it was some years ago. Nevertheless, the ultimate efficiency, expressed as brake horse-power hours per pound of fuel, will be affected by the lubricating oil, and its viscosity has been shown

by test to modify the fuel consumption on a brake horse-power basis by a small but definite percentage. In other words, best fuel consumption under test-bed conditions, which are the engine builders' guarantee conditions, could be obtained by using the lowest-viscosity oil which the tester dare put into the system, or one which having a steep viscosity-temperature relationship, loses viscosity rapidly as temperatures rise to working values. No consumption test, however, is complete which reports only on the fuel oil and disregards the lubricating oil, but unfortunately the duration of a fuel consumption test is generally too short for reliable lubricating oil results to be obtained. In the trunk piston engine, where cylinder lubrication is effected by oil thrown from the rotating parts, it is well known that lubricating oil consumption can vary between very wide limits, so that if on a comparative test the lower viscosity lubricating oil gave a reduction in fuel oil consumption of, say, 2·0 per cent at a cost of a 25 per cent increase in lubricating oil consumption there will not generally be achieved any true running economy.

In service the engineer's desire is to use an oil with the flattest viscosity-temperature relationship, the aim being to provide a lubricant, particularly for cylinder lubrication, which will give the safest and strongest oil film, always assuming that this desideratum is at least in part dependent on viscosity. Since, therefore, the modern engine allows considerable latitude in this respect, there is urgent necessity for recognition of minimum viscosities at elevated temperatures, the values being so chosen that the compromise between low viscosity for best economy and high viscosity for greatest safety is arrived at.

THE LUBRICATION OF STEAM LOCOMOTIVES ON THE GERMAN STATE RAILWAYS

By Professor Dr. H. Nordmann and J. Robrade*

To simplify the purchase and storage of oils, the policy of the German State Railways is to use as few grades of lubricants—and particularly special oils—as possible for its locomotives. The use of a number of different grades is unavoidable, however, owing to the number of points in a locomotive which require different treatment. In addition to points such as brakes, cocks, and valves, which require occasional lubrication, the following require attention:—

(1) Joints in driving rods and the valve motion, which consist mainly of hardened steel pins in bronze bushes.

(2) The bearings of the connecting rod and coupling rod, consisting of partly divided, partly closed bush bearings with gunmetal brasses and whitemetal linings, and hardened steel crankpins.

(3) The crosshead guide, of brass plates with or without whitemetal rubbing on a hardened steel guide.

(4) Axle bearings, of the same material as (2); the bearing brass embraces only part of the journal.

(5) The bearing of the extended piston rod, whitemetal on steel.

(6) The slide valve and cylinder, which must be steamtight at high temperatures.

The loading of the rod bearings is rather high; the pv values are 250–350 and the specific bearing loads p amount to 105 kg. per sq. cm. These values are possible as the bearings are cooled by their own movement and the rush of wind. The temperature rises to 40–50 deg. C., while the axle bearings, with similar pv values, show temperatures up to 80 deg. C. since they are shielded. The bearings of rods and driving gear are wick-lubricated, the axle bearings are lubricated through oil pads, and the driving gear receives lubricant in drops through the movement of the rod. The German State Railways have recently investigated wick lubrication and have found that the oil supply can be regulated by the size of the wick. The oil supply through the capillary action of the wick is reliable and does not cease suddenly as can happen with a mechanical supply when the oil is dirty. The wick acts as a filter and delivers clean oil. The disadvantages of wick lubrication are the reduction in the oil supply when the level in

* German State Railways.

the oil container sinks and that when the locomotive is not working the wicks must be drawn out to prevent loss of oil.

For cases (1) to (4) a medium heavy machine oil is used, which after service in summer or winter is separated and should satisfy the following requirements:—

(*a*) It must be free from solids, free mineral acids, free alkali, and thickening substances.
(*b*) The specific gravity at 20 deg. C. must not exceed 0·95.
(*c*) The open flash point should not be under 160 deg. C. for summer oil and 140 deg. C. for winter oil.
(*d*) The Engler viscosity at 50 deg. C. should be 8–10 for summer oil and 4·5–8 for winter oil.
(*e*) Hard asphalt should not exceed 0·2 per cent.
(*f*) Acid number not over 1·4.
(*g*) Water not over 0·2 per cent.
(*h*) Ash not exceeding 3 per cent.
(*j*) Summer oil must still be fluid at −5 deg. C., winter oil at −20 deg. C.

Oil answering to this specification is suitable for the lubrication of the driving gear and axle bearings of German locomotives. Only in the special case of high-speed locomotives is a better quality of oil desirable. In such cases, mixtures of summer and winter oils or light machine oil plus oil suitable for contact with wet steam are used to some extent and also special oils. Thus to lubricate a streamline locomotive on trial runs up to 200 km. per hr., oil to the following specification is used:—

Specific gravity at 20 deg. C., 0·915.
Open flash point, 215 deg. C.
Solidification point, below 20 deg. C.
Engler viscosity at 50 deg. C., 24.

In some cases, the so-called "Gleitöle" (Voltol oils) have been used successfully, but no improvement has been found to follow the use of graphitized oils. The problem of the passage of graphitized oils through wicks and pads is still unsolved.

Ordinary oil is thus used to lubricate the driving gear of locomotives on the German State Railways. A few trials have been made with lubricating greases, after making the necessary changes in the lubricant containers. Tests were carried out with a hard soda grease fed under spring pressure from grease cups into the bearing, and also with an oil and dust-proof bearing filled with the grease. These tests, which are not yet completed, served rather to decrease loss of oil and keep the locomotive clean externally than to improve the lubrication.

About 35 kg. of oil is consumed per 1,000 kilometres for the lubrica-

tion of the driving gear and axles of express locomotives, as against 18–25 kg. for passenger and goods locomotives. The oil is mainly lost through centrifugal action and dropping, so that discussion of the purification of oil used to lubricate steam locomotives is pointless.

As points (5) and (6) work in presence of steam they are jointly lubricated in the unit locomotives of the State Railways with cylinder oil by means of a pump; the pressure is 35 atmos., and 4 mm. piping is used. The supply at any point can be regulated. An oil seal is provided in the shape of a non-return valve which is actuated by the oil pressure and prevents either steam under pressure in the cylinder or slide valve from entering the oil line or the vacuum in the cylinder from sucking out the oil. Pumps have almost completely replaced the hydrostatic lubricators which were formerly used.

At points (5) and (6) the highest demands are made by the lubrication of the slide valve and cylinder, so that oil is used which conforms to the requirements of cylinder lubrication. Two types of oil are used: emulsified oil and superheated-steam cylinder oil. The former is prepared at various service centres on the line by mixing superheated steam cylinder oil with lime water as follows: 47 parts superheated steam cylinder oil, 3 parts Voltol oil, 50 parts lime water; or 43 parts superheated steam cylinder oil, 2 parts Voltol oil, 5 parts spindle oil, and 50 parts lime water. Emulsification tests were made on various superheated steam cylinder oils to ascertain their suitability, as the analytical data could not serve. Despite this, the following data are taken as characteristic for the basic oils:—

Specific gravity at 20 deg. C., 0·90.
Engler viscosity at 100 deg. C., 6·0–6·2.
Open flash point, 330–340 deg. C.
Acid number, zero.

The emulsion will only keep for a limited time and must not be exposed to high temperatures or to freezing. It is used in locomotives under medium loads instead of the usual superheated steam cylinder oil; the wear of cylinder and piston rings and the residue in the cylinder have been found to be not materially different from that observed with pure superheated steam cylinder oil.

Three grades of superheated steam cylinder oil are distinguished:—

(1) "Special high-quality cylinder oils" for locomotives with high superheat (up to 400 deg. C.) and high average piston speed (7–9 metres per second). Certain branded oils are admitted after physico-chemical examination and practical test. The analytical data are as follows:—

Specific gravity at 20 deg. C., 0·903.
Engler viscosity at 100 deg. C., 7·3.

Open flash point, 341 deg. C.
Acid number, zero.
Water, traces
Hard asphalt, 0·03 per cent.
Ash content, 0·03 per cent.
Colour, greenish.

(2) "High-quality superheated steam cylinder oils," also purchased by brands. The chief difference from oils of group (1) is that the flash point is lower. These oils are used in locomotives with high superheat but lower piston speed. Numerous practical observations have shown that the considerable frictional heat produced at high piston speeds through friction between piston rings and cylinder wall caused heavy demands on the cylinder oil, so that for locomotives with high superheat but a lower piston speed, an oil of lower flash point suffices.

(3) "Ordinary superheated steam cylinder oils" which are purchased by specification without special trial. The specification provides that these oils must be free from solid matter, free mineral acid and free alkali, and that they must not appear black by incident light.

Specific gravity at 20 deg. C., not over 0·95.
Open flash point, not under 300 deg. C.
Engler viscosity at 100 deg. C., 5–7.
Hard asphalt, not over 0·1 per cent.
Acid number, not over 0·7 per cent.
Ash, not over 0·10 per cent.
Moisture, not over 0·2 per cent.

On account of the importance of the special high-quality superheated steam cylinder oils for the express service, the tests are carried out according to a carefully planned record sheet, which shows the name of the oil and the supplier, the testing place, the locomotive used for the test, the number of tests, as well as time, distance covered, and performance of the locomotive in metric tons per kilometre, and the setting of the oil pump for the different lubrication points. Then follow data on the material and dimensions of the piston rings, and on the results of the physico-chemical examination of the oil. Records are made of the running load, the consumption of cylinder oil, and the loss of weight of the piston rings. For example, the appended results were obtained with a "special high-quality" superheated steam cylinder oil tested in a "series 03" express locomotive with a cylinder 570 mm. in diameter and five narrow piston rings after service over 42,000 kilometres:—

Oil consumption, 2·24 kg. per 1,000 kilometres (4·16 kg. per million locomotive service kilometres).

Piston rings, none stuck, none damaged; wear, 1·56 grammes per ring per 1,000 kilometres.
Cylinder bearing surface, smooth, polished.
Oil residues, insignificant.

The cylinder oil was evaluated as "very good" and accepted for service.

THE LUBRICATION OF MOTOR CAR ENGINES BY ADDITIONS TO THE CARBURANT

By Wa. Ostwald*

The lubrication of motor car engines can be accomplished by means of one of two distinct methods: (1) continuous-supply lubrication, and (2) fresh-oil lubrication. In the first method, a supply of oil is provided in the crankcase or elsewhere and the oil is pumped to the points requiring lubrication until, say, some 2,000 or 3,000 kilometres have been covered, the oil being replaced after having been used possibly 10,000 times. The method is especially suitable for use where the oil has also to serve as a cooling medium. The oil is evidently heavily stressed both thermally and chemically, and though oil coolers, filters, and crankcase ventilation provide some relief, they do not provide a cure.

In the second method, each lubrication point is supplied from time to time with the required small amount of fresh oil. This can be carried out by means of a metering pump or by mixing lubricant with the fuel. This method is the more natural and fundamentally better mode of lubrication. The oil suffers much less from thermal and chemical stress, but the chief point about this method is that the oil is used only as a lubricant and not as a cooling medium. A somewhat neglected type of fresh-oil lubrication, known as "mixture lubrication", consists in adding between 2 to 10 per cent of lubricating oil to the carburant. This procedure is often considered to be inferior, but it has proved its value for the lubrication of two-stroke engines in small cars and motor bicycles. Further it is to be noted that mixture lubrication has proved to be particularly good under the heavy strain of continuous running on the German State motor roads, whereas under unfavourable circumstances the usual method of lubrication by circulation allows the oil to attain inadmissibly high temperatures, so that special oil coolers are required. Mixture lubrication has also shown its value for outboard motors. It therefore seemed desirable to pay closer attention to a method which had proved itself in practice.

This wish is especially pertinent in view of the fact that after years of doubt, upper-cylinder lubrication, i.e. the addition of small amounts of special lubricants to the carburant, even for new engines which are not yet run in and are lubricated by circulating oil, has shown itself to be both practical and advantageous (diminution of the risk of piston seizure, reduction of wear due to cold starting, easier starting, the

* Wirtschaftsgruppe Kraftstoffindustrie.

prevention of "Ricardo corrosion" and so on). In fact, mixture lubrication is very simple whether from the point of view of construction or of operation. Viscosity index, oxidation resistance, and other properties which are essential for circulating oil are of no account when this method is used. Even the usual warming-up of the engine until the oil begins to circulate is no longer required. Up to the present, mixture lubrication has been held to be not only of little utility but wasteful. To-day the contrary appears to be true, at least for two-stroke engines.

The Rubbing Surfaces. The shape and nature of the rubbing surfaces are of importance. An advantageous shape is one which, by its own pumping action favours the formation of a bearing oil wedge or oil cushion and increases the load-carrying capacity of the oil film. It is important to note that in mixture lubrication, even with known bearing temperatures and a lubricant of known characteristics, a definite viscosity of the oil cannot be assumed since, as a rule, the extent to which the lubricant is separated from the particles of carburant is not known. In general it must be assumed that the separation of the high-boiling fractions of the fuel is incomplete, so that the oil used as lubricant can have a lower viscosity.

Comment is not required on the physical nature of the sliding or rolling surfaces in respect of mixture lubrication as compared with ordinary lubrication. The conditions of dry friction are just as significant for mixture lubrication as for any other method of lubrication because they come into play as soon as dry friction occurs. In mixture lubrication, however, there is less danger of failure, particularly of piston lubrication on starting from the cold, than with circulation lubrication. Boundary friction is, so to speak, the rule in the motor car engine, particularly in the cylinder, and this applies equally to mixture lubrication as well as circulation lubrication.

Incomplete lubrication comes about partly because the layer of lubricant does not adhere firmly enough to the surface and thus is removed, for instance, by condensed steam of combustion, or because the layer, chiefly at high temperature, is not sufficiently resistant to pressure, so that metallic contact and dry friction occur. Attempts have been made to render the oil film more resistant to stress. Its adherence to the metal has been improved by adding chemicals to the oil, particularly for "running-in oils" for circulation lubrication, a method which is probably more effective with mixture lubrication as the added material arrives at the point of lubrication in a fresh condition. A similar effect is attributed to graphitized surfaces.

The selection of oil stocks, the use of special methods of refining,

the addition of acids ("Germ" oils), or sulphur, lead, and chlorine compounds (as in extreme-pressure lubricants), have all been employed in order to make the oil or some of its constituents corrode the metal slightly and so anchor the oil molecules on the surface. It can also be assumed that such a weak corrosion will leave the rubbing surfaces with extremely minute roughnesses so that the oil film will be physically more resistant to removal. With extreme-pressure lubricants, it appears that the weak corrosion of the bearing surfaces leads to the formation of thin films of compounds such as ferrous sulphide, iron oxychloride, etc., which offer physical resistance to the removal of the oil film and also prevent or hinder metal-to-metal contact and therefore local welding and seizure. Iron or light-alloy pistons with very thin coatings of zinc or cadmium are also used for the same purpose, though these coatings may have an influence on the clearance. Aluminium pistons covered with a thin skin of porous aluminium oxide ("Eloxal" process), are also used to the same end, while a similar effect is given by colloidal graphite added to the oil, lead compounds, and chloro- and bromo- compounds derived from "leaded" petrol, and finely divided iron oxide derived from fuels containing iron carbonyl.

Influence of the Carburant. The influence of the fuel on lubrication appears, somewhat unexpectedly, to be smaller in mixture lubrication than in circulation lubrication. This is because the circulating oil contains combustion residues, whereas with mixture lubrication fresh oil is continually brought in by the fuel. Both methods of lubrication are affected by oil dilution due to high-boiling fuel residues. In circulation lubrication the extent of oil dilution depends upon the type of carburetter, the working temperature, the oil temperature, the ventilation of the crankcase, etc., as well as on the boiling range of the fuel. Consequently, oil dilution varies considerably according to working conditions.

It is not known to what extent high-boiling constituents of the fuel have an influence on mixture lubrication. As very fluid oils are successfully used in mixture lubrication, or, in other words the use of viscous oils is neither necessary nor desirable for this type of lubrication, the influence of the high-boiling constituents would appear at least to be no greater than in circulation lubrication. This is somewhat surprising when it is considered that petrol now contains fractions boiling at up to 200 deg. C., whereas the temperature usually does not exceed 100 deg. C. in the crankcase of a two-stroke engine, i.e. in the place where the fuel and oil chiefly separate. It might thus be supposed that in mixture lubrication a solution of the oil in the high-boiling constituents of the fuel is employed. For parts which run cold that

would be advantageous, as is shown by the common use (e.g. the Ford method) of very fluid oils (4 deg. Engler instead of 12 deg. Engler). It may be of importance that, in mixture lubrication, the diluting constituents evaporate at hot points, the oil content of the lubricant thus adjusting itself almost automatically to the temperature of the lubricating points.

The different constituents of the fuel have varying effects on lubrication. Thus benzol and especially alcohol are slightly detrimental to lubrication. The effect of benzol is probably related to its capacity for removing grease, as a steel rod washed in petrol rusts less quickly than one which has been washed in benzol. Alcohol probably acts in a similar way.

Water is detrimental to lubrication, whether it comes from the cooling system, or from condensation when the temperature of the combustion gases falls below the dew point (Ricardo corrosion). This detrimental effect must be mainly due to evaporation of the oil film, just as the chemist cleans oily vessels by means of condensing steam.

The fuel has a decidedly favourable influence on lubrication when it contains lead compounds and iron carbonyl, though the mechanism of the effect still requires to be explained. The effect does take place and it has led to the so-called "upper-cylinder" lubricants, which are materials added in low concentration to the fuel in order to improve lubrication.

Upper-Cylinder Lubrication. In mixture lubrication all the lubricant required is mixed with the fuel, whereas in upper-cylinder lubrication, lubrication is usually by means of circulating oil and some 0·1 per cent of upper-cylinder lubricant is added to the fuel in order to facilitate starting from the cold, the lubrication of the inlet and exhaust valves, and the removal of carbon deposits, etc. Between these two extremes lie other cases. Thus in two-stroke engines with circulation lubrication, part of the lubricating oil is added to the fuel. For four-stroke engines with circulation lubrication which tend to seize when new or after repair, it has long been usual to add a little lubricant or machine oil to the fuel.

The efficacy of upper-cylinder lubricants was for long doubtful, partly because a sufficient distinction was not made between products designed to give known effects and secret materials for which quite astonishing claims were made, such as 30 per cent economy in fuel and so on. In Germany, thanks to a regulation of the "Überwachungsstelle für Mineralöl", the claims made for such secret materials must be substantiated in a State Institute before the materials can be sold. As a result all these secret materials suddenly disappeared from the German

market, thus marking the culmination of a campaign which the author had waged for over ten years.

It is necessary to differentiate between upper-cylinder lubricants for four-stroke engines and those for two-stroke engines. In most engines the stems of the inlet and exhaust valves are not included in the lubrication system, so that the upper-cylinder lubricant has an important task to perform. The stems of the inlet valves are not very hot, so that, although the mixture of fuel and air has not time to arrive at equilibrium and is still partly in the form of a mist, only fuel particles carrying very liquid fractions of the upper-cylinder lubricant can be effective. Despite this, the lubrication of inlet valve stems in this way has proved its efficiency in practice.

The lubrication of the exhaust valve is particularly difficult, since here it is expected that at least a part of the very small addition made to the fuel will escape combustion and, leaving the combustion chamber with the burnt gas, will deposit itself on the exhaust valve stem. Although most German engines run on a reducing mixture and only have to deal with a local excess of oxygen, this behaviour of an oil, present to the extent of only some 0·1 per cent in the fuel, is remarkable. The method, however, has again shown its value in practice. Similarly, most of the claims made for upper-cylinder lubrication, such as easier starting from the cold, the softening of the hard and partly asphaltic oil near the piston rings, and the dissolution or diminution of the carbon deposit, have been proved correct in practice.

The question remains to be answered whether upper-cylinder lubrication is worth while or not. The author knows of no experience in this respect which is not open to objection. From experience of circulation lubrication it might be thought that with mixture lubrication upper-cylinder lubrication would be useful, and this naturally suggests that the effective constituents of the upper-cylinder lubricant should be added from the beginning to the lubricant employed.

What are the effective constituents of the upper-cylinder lubricant? The basis is a very thin mineral oil, which thus has a high ignition point, evaporates easily, and forms carbon when cracked. Solvents for asphaltic materials (e.g. tetralin) are also added, other materials being used which improve the wetting capacity or act like extreme-pressure lubricants and produce very mild corrosion. Others, like camphor or naphthalene, are supposed to prevent oxidation or polymerization. Various aromatic amines both prevent oxidation and fix acid, whilst other bodies, such as hydrogenated alcohols, have an emulsifying effect on the water of combustion produced on starting from the cold. Colloidal graphite is also used as an upper-cylinder lubricant, though it is difficult to maintain in dispersion in the fuel. The very fine

coagulum, which tends to float, causes no trouble, however, in the filter or carburettor.

Upper-cylinder lubrication can only affect the driving mechanism in so far as upper-cylinder lubricants are present in the circulating oil.

Practical Applications. In Germany mixture lubrication is applied exclusively to two-stroke engines, large numbers of which are lubricated in this way. In early days a mixture containing 10 per cent of oil was used, though now 4 per cent is more usual, and even weaker mixtures are used, the concentration used for run-in engines being often reduced to 2·5 per cent. More than 4 per cent of oil is rarely used except for new engines. It is not usual to add upper-cylinder lubricants to the mixture. The cost of the ordinary mixture containing 4 volumes per cent of oil may seem high, when it is compared with that of circulation lubrication, as the consumption in new engines is often insignificant for months and may even become apparently negative owing to oil dilution. But when the replacement of circulating oil is considered, then the consumption of oil in mixture lubrication appears by no means too high, being about 4 per cent compared with circulation lubrication, though it should be 2·5 per cent, or even less than the consumption, including replacements, which is usual with circulation lubrication. However, circulated oil can be regenerated to advantage, whereas in mixture lubrication the oil is completely lost, though the economics of oil regeneration is not very significant as regards motor fuel.

The limits of the concentration of oil in mixture lubrication are not sharply defined. They depend largely both on the mechanical condition of the engine and on the nature of the oil. It is particularly important to note that mixture lubrication can only be applied where the lubricant has not to act at the same time as a cooling fluid; this is why roller bearings are preferred when mixture lubrication is employed.

The use of small additions of oil in the mixture lubrication of two-stroke engines is not hindered, in the presence of easily lubricated roller bearings, by the necessity for adequate lubrication of the piston. Surprisingly low amounts of oil are found on cylinders, pistons and piston rings in good order, without any noteworthy wear or damage resulting. Even comparatively worn, leaky pistons hardly cause any trouble with mixture lubrication, though the oil consumption does not increase at all. It is interesting to note that two-stroke engines begin to smoke as soon as the pistons begin to allow blow-by, whereas the same engine with gastight pistons, the same oil concentration, and the same oil consumption does not smoke.

The use of smaller proportions of oil is suggested by the danger that

the gas slots may be blocked with carbon and that the piston rings may seize. With 4 per cent and less of oil modern two-stroke engines practically do not require decarbonization, so that it is uncertain how far the effect is due to modern gas control with flat pistons or to modern lubricants. That the former plays an important part is shown by the fact that existing machines with certain types of pistons still often show deposits of carbon.

For two-stroke engines pure mineral oils of not too high a viscosity are advisable. Certain special oils which, owing to their low viscosity, mix easily with petrol, are not prepared by pre-dilution but by the choice of fractions of low viscosity. Heavily compounded oils are inadvisable owing to the danger of the formation of carbon. Non-mineral oils are unsuitable because of their low solubility in the usual fuels. The addition of alcohol reduces the solubility of pure mineral oil in the fuel almost to the limit, chiefly in winter, but no harm appears to result.

Mixture lubrication can be conceived as a measured fresh-oil lubrication since it brings fresh oil into use in proportion to the fuel consumption. As a result the fuel is distilled off from the oil, so that a continuous supply of oil is provided in the crankcase. The extent to which the oil separates from the mixture of fuel and air in the crankcase is not known, neither is the extent to which it arrives as a mist in the combustion chamber, in part being burnt, in part being deposited on the cylinder walls. Both of these suppositions appear to be significant because a supply of oil is always found in the crankcase when the engine is dismantled, and because mixture lubrication never fails even in the greatest cold, when oil can hardly be distilled off in the crankcase.

Mixture lubrication cannot become usual so easily in four-stroke engines, though it has great advantages, as it renders the oil pump, pipework, etc., unnecessary. The reason does not lie so much in the fact that in four-stroke engines the mixture of fuel and air does not pass through the crankcase,* as in the practical experience that the mixture of oil and fuel easily leads to deposits on the inlet valves. However, mixture lubrication in the two-stroke engine is so very simple, cheap, and sure that it can hardly be replaced.

Conclusion. Engine lubrication stands, so to say, at the crossroads. The increasing improvement in production methods has made it possible to consider the lubricant in an engine with circulation lubrication almost as a part like a sparking plug or a piston, except that

* With mixture lubrication oil can pass from the combustion chamber into the crankcase.

oil and sparking plug are changed comparatively often, piston but rarely.

With mixture lubrication, however, the total consumption of lubricant is proportional to the fuel consumption.

If circulation lubrication appears to be superior, the contrary opinion will be arrived at if the matter is considered from the technical point of view. The sole fact that mixture lubrication always withstands running on special motor roads, whereas, for equal reliability, circulation lubrication requires the provision of an oil cooler and a thermostat, shows the technical superiority of mixture lubrication as of any system of controlled lubrication with fresh oil. From the economic standpoint, considering the low concentration of oil used in mixture lubrication, circulation lubrication should lose its superiority as the regeneration of the used oil, which is entirely lost in mixed lubrication, plays no part.

When it is considered that mixture lubrication is applied in a comparatively less widespread type of engine, whereas circulation lubrication can look back on a very wide and thorough development, there would appear to be good prospects for the improvement of its standing.

THE PROBLEM OF OIL CONSUMPTION IN HIGH-SPEED INTERNAL COMBUSTION ENGINES

By E. C. Ottaway*

The problem of oil consumption in high-speed internal combustion engines has been of prime importance for many years to those responsible for the cost of operation of such units and, in consequence, has received much attention. The problem, however, is still of major importance to the maintenance engineer, indicating that the work which has so far been carried out, whilst excellent within its scope, has not completely covered the ground. From time to time claims are made that a panacea for this evil has been discovered, but it is usually found that the discovery is a cure only in certain circumstances.

The oil consumption characteristics of an engine may be classified broadly under two headings: the initial oil consumption and the maintenance of a satisfactory oil consumption over an extended mileage. The former of these characteristics places a given design of engine within a certain category as to its oil consumption possibilities, the latter largely depends upon wear.

The second characteristic has mostly been the subject of research and it appears possible to obtain satisfactory results within the limits imposed by the first characteristic, in regard to which the author contends that little is known. Certain very clear indications, however, arise from the study of service experiments carried out on large numbers of engines, which serve to indicate the lines on which future research might profitably be undertaken.

The limiting factor of engine life for road vehicle operation is lubricating oil consumption, since the engine must be removed when the consumption reaches such an amount that objectionable exhaust gas is produced or, alternatively, road troubles are introduced by fouling of the sparking plugs; it is rare that the overhaul of an engine is warranted by reason of the actual cost of the oil consumed.

To simplify discussion, the factors governing the oil consumption of an engine are classified under the following headings: (1) combustion characteristics; (2) mechanical design of engine; (3) design of piston (including arrangement and design of rings); and (4) wear of pistons, cylinders and rings. The initial oil consumption is affected by items 1, 2, and 3, and the maintenance of oil consumption by the whole four.

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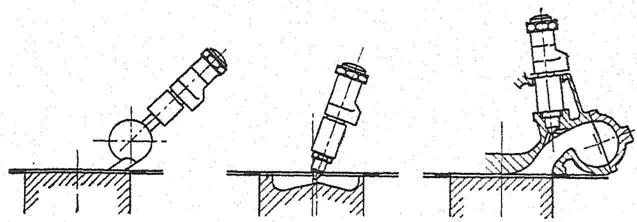
Combustion Characteristics. The effect of the combustion characteristics on the lubricating oil consumption was first drawn to the author's attention by service experience with various makes of compression-ignition engines. It became apparent that the effect of this factor was so marked as to divide engines into two distinct primary categories. The effect is such that in an engine, of which the combustion characteristics are favourable to a low oil consumption, the problem hardly exists to-day, provided that full advantage is taken of knowledge available for the choice of materials for the cylinder bore, and the design of the piston. Conversely, where the combustion characteristics are unfavourable to a low oil consumption, the problem is still very real, and cannot be solved by the many known



a *b* *c*

Fig. 1. Types of Cylinder Head

a Open chamber. *b* Indirect or ante-chamber. *c* Compromise design.

expedients. The author is aware that this theory has not so far received much consideration. He believes, however, that the remaining problems of oil consumption can be solved if the mechanics of the phenomena in relation to the combustion characteristics are fully understood. A study of service results with varying types of engine operating on what may be broadly classed as the direct and indirect cycles of operation, clearly indicates the superiority of the former in respect of oil consumption. The matter has also been studied by comparing the results obtained from a given mechanical design of engine of approximately 7 litres capacity, operating on three different characteristics, the only variations to the engine being the changes in the cylinder head and piston crown required to provide the differences in combustion characteristic. The three types of cylinder head used are illustrated in Fig. 1.

The average oil consumption for the first 3,000 miles and at 30,000 miles, are given in Table 1. The results are sufficiently definite to confirm the opinions expressed above.

Consideration of this problem suggests that it may be related to the turbulence within the cylinder, serving to remove the lubricating oil

TABLE 1. EFFECT OF COMBUSTION CHARACTERISTICS ON OIL CONSUMPTION

Mileage stage	No. 1	No. 2	No. 3
3,000 miles . .	536	6,700	4,922
30,000 miles . .	200	1,000	250

left on the cylinder walls. Ricardo has investigated the effect of turbulence on lubricating oil consumption, but his findings were of a somewhat negative character in that a direct conclusion was drawn that such connexion did not exist, though a rider to his conclusions indicated that cases were known where swirl had been found to affect the rate of lubricating oil consumption.

Service experiments are necessarily prone to confusion by many variables, but it is possible to discern a general tendency to increasing oil consumption with the introduction of engines of high turbulence, although the data are much clouded by other changes and by increases in speed and specific output.

TABLE 2. AVERAGE CONSUMPTION OF FUEL AND LUBRICATING OIL BY SIX ENGINES

	Before fitting new pistons, average mileage per gallon	After fitting new pistons, average mileage per gallon
Fuel	8·88	8·12
Lubricating oil . . .	105	2,290
Fuel and lubricating oil .	8·18	8·09

A further fact which supports the relationship postulated is that an increase in lubricating oil consumption, particularly on the indirect type of compression-ignition engine, is accompanied by a corresponding decrease in fuel consumption. In fact, in assessing the merits of various engines in respect of fuel consumption, a correction has to be made for this factor. As an illustration, Table 2 shows the average fuel and lubricating oil consumption of six engines for a period of approximately 3,000 miles before and after fitting new pistons. It will be seen that the lubricating oil consumption has greatly improved, but

that this improvement is accompanied by an increase in fuel consumption, the combined figure remaining practically constant. This relationship is confirmed by observation of large numbers of engines.

That this characteristic should be so marked is a clear indication that the lubricating oil is being burnt with some degree of efficiency, and this is consistent with the fact that the compression-ignition engine is able to run at very much increased oil consumptions, compared with a petrol engine, before producing objectionable exhaust smoke.

General Mechanical Design. The general mechanical design of the engine has a considerable bearing on oil consumption, in so far as it affects the amount of lubricating oil thrown on to the cylinder walls. Researches carried out by Ricardo have indicated that, other factors being equal, the consumption of lubricating oil is roughly proportional to the amount of oil thrown on to the cylinder walls. It is clear that an engine where this amount of oil is originally excessive will tend to show a poor initial oil consumption or, alternatively, an engine which, by reason of rapid mechanical wear permits a large increase in the amount of oil thrown on to the cylinder walls, will exhibit a poor maintenance of oil consumption.

Experience suggests that this aspect of the problem has been considerably aggravated in the compression-ignition engine by the enforced use on many engines of bearing metals requiring high initial running clearance. In order that a rapid deterioration of oil consumption from this cause may be obviated, attention should be paid to the width of main bearings, particularly the intermediate bearing on six- and eight-cylinder engines. It has been the practice to make these bearings somewhat narrow, and whilst complying with requirements in respect of load factor, sufficient resistance to the passage of oil from the bearing is often not provided, particularly when a narrow bearing of this type is used in a crankcase which is prone to excessive distortion under load. Under these conditions, by reason of crankshaft and crankcase distortion, a barrel shape is produced on the bearing, providing an easy passage for oil.

Many attempts have been made to control the splash of oil inside the engine so as to prevent an undue amount reaching the cylinder walls. The author's experience is that these attempts have invariably failed, and consideration of conditions inside the crankcase in respect of the amount of oil present, and the continual rapid movement of air and oil mist, does not lead to an optimistic view of the ultimate success of such expedients.

Design of Piston. The major problem in piston design is to operate

the piston at the minimum practical clearance without susceptibility to an undue proportion of seizures on test. At first sight, the cast iron piston would appear to offer some advantage in this direction. In practice, however, this advantage is generally offset by a greater tendency to seizure and the more extensive nature of the resulting damage. This point is of particular importance where engines must operate on full load with a relatively short running-in period.

The second item affecting cylinder wear and thereby lubricating oil consumption is the depth and clearance of the top piston-ring land. Experiments carried out on many engines both of the petrol and compression-ignition type, have repeatedly demonstrated the importance of this factor. It would appear that a narrow top land or a top land with excessive clearance will provide too easy a passage for the pressure in the cylinder to reach the area behind the piston ring with a consequent artificial increase in ring pressure. Conversely, where the depth of the land is ample and the clearance small, the wiredrawing effect would appear to be sufficient to reduce considerably the pressures behind the ring.

The piston should be as rigid as possible, since breathing of the casting will adversely affect lubricating oil consumption. Many pistons of what may be termed the "self-correcting" type have a notoriously bad effect on lubricating oil consumption when fitted to engines of over $3\frac{1}{2}$ inches bore. Apparently the movement of the piston skirt has some bearing on the mechanical conveyance of the lubricating oil from the lower part of the cylinder to the combustion chamber. The author does not attempt to deal further with this fact since the mechanism of this transference of oil seems to be but imperfectly understood.

In considering the design and arrangement of piston rings, one is faced with many conflicting desiderata. The general tendency in commercial vehicle engines during recent years has been to increase the number of piston rings and to employ at least two of the special high-pressure scraping type.

In regard to the design of the pressure rings, the author's experience is that a wide top ring with consequent low cylinder wear is of more advantage than a narrow high-pressure ring. The latter will produce unusually good oil consumption during the initial stages, but by reason of the increased cylinder wear occasioned by this arrangement, the increase in oil consumption is rapid.

Much has been written recently about the general results to be obtained from the use of high-pressure rings. One authority claims that, by the use of sufficiently high-pressure rings, and an ample supply of lubricant to the cylinder bore, the problem is solved. In the

author's experience the solution is not so easy. It will be generally accepted that the maximum cylinder wear occurs opposite the pressure ring belt and is greatest in the region of the position of the top piston ring when the piston is at top dead centre. The wear of the lower half of the cylinder is negligible. It is difficult to see, therefore, why a copious supply of lubricant to be scraped off by the pressure rings, should in any way alleviate this condition. Service experiments carried out with rings of varying pressures have indicated that no advantage is to be gained in overall oil consumption over a prolonged period of mileage.

Wear of Pistons, Cylinders, and Rings. An increase of lubricating oil consumption is caused by wear of the piston rings, the ring grooves and the cylinder bore, and, to a lesser degree, the piston. Wear of the ring grooves largely depends upon the rate of cylinder wear, since groove wear is caused by transverse movement of the ring in the slot produced by the irregular nature of the wear on the cylinder. If the rings are manufactured from satisfactory material, the rate of wear of this part will be low, provided that the cylinder wear is small. The author contends that the important factor to consider from the wear standpoint is the primary one of the cylinders themselves.

The subject of cylinder wear has been dealt with most ably by C. G. Williams, and congratulation is due in regard to the manner in which the researches sponsored by the Institution of Automobile Engineers have been carried out. The corrosion theory, which has been the outcome of these researches, has been criticized because it does not explain all the factors. The author does not think that it was ever intended that it should. There is no doubt, however, in the author's mind that the corrosion theory is the correct explanation for part of the wear on the cylinder. The exact proportion depends logically on the condition of operation, and is high where operation is intermittent. Service tests have been carried out in order to check the research results. Many materials have been tried, but it will suffice to deal with those which have a direct bearing on the matter. Three particular specifications are referred to together with the relative rates of wear (Table 3).

The last named of these materials is of an austenitic type, and, in accordance with the corrosion theory, gave the greatest improvement. Moreover, results of previous tests indicate that analyses 1 and 3 are substantially equal in resistance to abrasive wear. The test results were obtained on buses operating in central London, with frequent stops. The rate of cylinder wear obtained with the first material was considered to be good in relation to the operating conditions, and was

obtained with the material in the hardened condition. The second material was tested in the condition as cast, and this again showed some substantial improvement. The difference, in this instance, might well be an approximation to the amount which may be obtained by an improvement in resistance to abrasive wear.

The author's experience over a number of years is that the important item in a cylinder iron of the non-austenitic type is the phosphorus content. This fact, it is believed, is now being generally accepted. Much rapid cylinder wear has undoubtedly been occasioned in the past by the use of a cylinder iron which is almost lacking in phosphorus, and has, in consequence, given very rapid wear, although possessing all the virtues in respect of machinability and foundry use. Under similar conditions to those on which the above materials were tested, a cylinder iron conforming to material No. 1, but lacking in phosphorus, shows a rate of wear as high as 0·0005 inch per 1,000 miles.

TABLE 3. WEAR OF DIFFERENT CYLINDER MATERIALS

Analyses	1	2	3
Total carbon, per cent.	3·25	3·13	2·75
Combined carbon ,,	0·65	0·54	0·59
Silicon ,,	2·05	2·335	1·73
Manganese ,,	0·90	0·75	0·69
Sulphur ,,	Not over 0·12	0·83	0·065
Phosphorus ,,	0·65	1·26	0·707
Nickel ,,	0·37	0·37	11·565
Chromium ,,	0·20	0·07	2·635
Copper ,,	—	—	4·620
Rate of cylinder bore wear, inches per 1,000 miles	0·000316	0·000262	0·000182

Conclusions. These brief notes constitute an opinion rather than a proof, which has been arrived at as a result of many service tests and the accumulation of data from the operation of some 6,300 vehicles. An attempt has also been made to indicate the extent to which the researches carried out during the past few years have assisted the maintenance engineer and the lines on which future research work is required. To summarize this position, it may be stated that the problems associated with the wear of the piston and cylinder assembly have been well covered, it being the author's contention that by the judicious use

of the materials available to-day the wear can be kept to a satisfactory and even creditable figure. There is, however, considerable opportunity for research into the reason for which some engines show a satisfactory initial oil consumption, whereas others will yield to no treatment. The solution appears to lie somewhere in the study of the effect of the combustion characteristics on oil consumption. This aspect of the problem may well be considered "the missing link".

THE WEAR OF CAST IRON

By J. G. Pearce, M.Sc.*

The general suitability and use of cast iron for steam and internal combustion engine cylinders, liners, and piston rings has led to much investigation of the factors governing wear, for a time with inconclusive results. In recent years a marked change has taken place, followed by material improvement in service life. The problem of wear arises mainly in the internal combustion engine, although the conclusions reached apply to other cases of wear in which lubricated grey cast iron is used.

Wear Tests. Wear resistance is not a specific property. It depends upon a variety of test conditions, and when determined, is strictly valid only for those conditions. In testing wear, laboratory apparatus designed to work under simplified conditions gives results quickly, but not necessarily similar to those obtained under actual service conditions. At the other extreme, service tests in vehicles are difficult to supervise and take time to furnish results, which are influenced by factors extraneous to the material such as design, lubrication, the presence of dust, and operating difficulties and conditions. This has led to the use of engines running in the laboratory under controlled conditions, but closely simulating those found in service.

Early tests on reciprocating wear omitted to take into account the fact that wear is mutual between two surfaces, the wear on the test piece depending in part upon the material against which it is worn. Surface condition is also important, and in general the harder the components and the less the wear between them, the more important does a smooth initial surface finish become. With due regard to these points, the wear of cast iron is mainly governed by structure and composition. Load and temperature within the ordinary limits of engine operation do not appear to be important. It is agreed that there is no general relation between wear and hardness, or machinability. For materials of the same structural type, however, increase in hardness may be expected to result in increased wear resistance.

Effect of Composition. Grey cast iron is essentially an alloy of iron and carbon, containing carbon in excess of that present in the steels. This excess is normally converted by silicon, which goes into solution,

* British Cast Iron Research Association.

from the massive and hard, brittle iron carbide or cementite to iron and graphite, the amount of silicon required depending upon the section and graphite, the amount of silicon required depending upon the section, slowly cooled thicker sections requiring less than thinner sections and having heavier graphite flakes. The small amount of sulphur present is usually fully neutralized by manganese and is present as occasional crystals of manganese sulphide. Any excess of manganese is present with iron carbide as manganese carbide. The only other element normally present is phosphorus, which forms a hard brittle phosphide eutectic in proportions depending upon the amount of phosphorus present. The carbon (usually up to about 0·7 or 0·8 per cent) not present as graphite exists as iron carbide, in a duplex or laminated structure along with iron (the ferrite of the metallurgist), the whole comprising pearlite. If carbon above 0·7 or 0·8 per cent did not exist as graphite, it would be present as massive carbide, making the metal virtually unmachinable. The task of the founder is so to adjust silicon content and section that enough carbon remains combined to yield a fully pearlitic casting, avoiding at one extreme either the presence of massive carbide, which is hard, or at the other, the conversion of pearlite in part to ferrite, which is soft. The permissible limits of variation of carbon and silicon in castings of given section are well defined, and small variations do not appreciably influence wear resistance. It is generally agreed, however, that increased silicon reduces wear resistance. Although used primarily for its action in breaking down carbide to iron and graphite, silicon has intrinsically a hardening and embrittling action. The effect of phosphorus in cast iron on wear has been much studied. The hard phosphide eutectic, in relief in a softer matrix, has the effect of the hard constituent of a bearing metal with respect to wear, and it is generally agreed that phosphorus increases wear resistance. Thus it can with advantage be used to the limit imposed by other considerations, a limit higher for large and, or alternatively, simple castings than for those which are small or complicated or both. The effect of graphite is less certain. Little weight is now attached to the so-called lubricating effect of graphite *per se* in cast iron, although the cavities may act as minute oil containers. Experiment and practical experience alike have shown that cast iron structures containing flake graphite wear well, while the very finely divided eutectic or pseudo-eutectic graphite has been found generally to be bad for wear. Until further work is done it is uncertain whether this result is characteristic of all structures in which fine graphite appears and why, and what the optimum size is.

Alloy additions are freely used, and include nickel, chromium, copper, molybdenum, titanium, and vanadium. Those having a

graphitizing action are useful in partially replacing silicon, giving a tougher matrix, while those having carbide-stabilizing or hardening action act in hardening the matrix, raising the combined carbon and ensuring a fully pearlitic structure.

Effect of Structure. In spite of its duplex structure, pearlite within the ordinary limits of engine operating temperatures behaves as a simple material. Normal cast iron, at one time a mixed ferrite-pearlite structure, is now as a rule fully pearlitic, interrupted only by graphite and such phosphide as may be present intentionally. Investigation has shown that a pearlitic iron is more resistant to wear than a ferritic iron or iron of mixed ferrite-pearlite structure. Wear between moving parts is at a minimum when there is at least difference in hardness between them, and this minimum is lower for pearlitic than for ferritic structures. It appears to be lowest when the moving part is the harder.

The work of the Research Department of the Institution of Automobile Engineers has been of great value not only in dispelling erroneous ideas about factors contributing to wear, such as petrol-diluted lubricants, rich mixtures, delayed lubrication, and elevated engine temperatures, but also in directing attention to the components of reciprocating wear in internal combustion engines (corrosion and abrasion), and has enabled wear to be looked at from a new point of view, particularly with regard to the effect of corrosion. It is well known that cylinder wear, confined to the piston ring track, reaches a peak at the combustion end. The corrosion effect takes place on starting when the cylinder wall is cold and receives moisture condensed from burnt fuel, and, of course, tends to predominate in engines required to start frequently, especially when restarted from cold. The abrasion effect predominates in engines running for long periods at comparatively high temperatures. This work gave a fresh stimulus to trials of austenitic cast irons, two of which are available, "Ni-resist" and "Nicrosilal" for cylinder liners. These irons have a structure totally different from that of pearlitic cast irons. When heated above 740–780 deg. C., a pearlitic iron begins to take graphite into solution as iron carbide, up to a limit at about 1,150 deg. C. of 1·7 per cent carbon, forming the austenite of the metallurgist. Certain elements have the power to depress the temperature at which this change begins (normally about 750 deg. C.) and to reduce it below atmospheric temperature. The metal is then austenitic in the cold state and consists of austenite with graphite, and, where chromium is present, of small amounts of chromium carbide. The austenitic irons are markedly different from the pearlitic irons in their relatively high ductility, softness, and resistance to corrosion. They are also non-

magnetic and are not susceptible to heat treatment. They have shown striking wear resistance in actual service both for corrosive and abrasive conditions.

Thermal Treatment. The thermal treatment of pearlitic irons resulting in structural changes (for example, from pearlite to ferrite, as by ordinary annealing) is detrimental to wear resistance, although such treatment involving no structural change does not have this effect. It has been fully demonstrated that the heat treatment of pearlitic irons promotes resistance to wear. The development of nitrogen-hardened cast irons and of chromium-plated irons has also made available other types of finish to resist wear, and these, in a number of cases, have given remarkable results. Martensitic irons can be produced either by alloy additions or by the thermal treatment of pearlitic irons, but experience varies as to their wear-resisting properties, which depend to some extent on conditions of operation. As indicated above, the more wear-resistant the various parts become, the more important becomes a smooth initial surface. In a comparatively soft ferritic matrix, capable of a greater measure of plastic distortion, and of taking a surface glaze, such finish is less important. The report of the earliest investigation undertaken by the British Cast Iron Research Association, on cylinder blocks, urged the adoption of the separate liner as a solution to the wear problem. Their adoption has been greatly aided by the development of the centrifugally cast liner, employing alloy additions and lending itself to thermal or surface treatment.

Conclusion. The developments briefly described have been such that, while the wear problem can by no means be described as solved, it has ceased to be the major problem of the internal combustion engine designer.

FILTRATION AS AN AID TO LUBRICATION

By J. A. Pickard*

The efficiency of lubrication of the moving parts of machinery depends in the first place upon the nature of the lubricating oil used, but the efficiency of the lubricating oil will be greatly impaired or even nullified if abrasive matter is permitted to enter it and to circulate to the moving parts with it. The various means which may be adopted to keep abrasive matter out of lubricating oil fall under the following heads.

(1) *Draining and Refilling*. If the whole of the oil in the system is removed from time to time and replaced by fresh oil, practically the whole of the suspended matter will be removed with it; although there will be some slight tendency for particles which have settled out not to be flushed out with the oil on draining and to be mixed in again with the fresh charge of lubricating oil when it is introduced. In this procedure abrasive matter is not removed from the oil, but is prevented from exceeding any given concentration.

(2) *Filtration*. Filtration may be applied in various ways. For instance, it may be applied to the whole of the oil going to the bearings ("full-flow filtration"), and this in turn may be filtered (*a*) finely, or (*b*) coarsely. Alternatively, only a part of the oil going to the bearings may be filtered ("bypass filtration") either finely or coarsely. In this procedure abrasive matter is removed from the oil during service and any desired degree of cleanness may be produced.

(3) *Centrifugal Action*. The accentuation by means of centrifugal force of the tendency of particles to settle may be usefully employed. In this procedure abrasive matter is removed from oil during service, but the degree of purification possible is less than with filters.

Nature of Suspended Material. The suspended matter in lubricating oil, especially that employed in internal combustion engines, is of various kinds, of which some are harmful, while the bad effect of other sorts is not so generally admitted. The material will consist of:—

(*a*) Metal particles arising from scraping, filing, and machining in the construction of the engine. These naturally will be present to a greater extent, if not solely, in a new or reconditioned engine.

(*b*) Particles which arise from wear in the engine.

(*c*) Sandy and gritty particles which gradually detach themselves

* The Metafiltration Company, Ltd.

from castings or which may enter the system from the air, either through the induction or by casual settling.

(*d*) Carbon. This may be present in almost any degree of coarseness. In some high-speed compression-ignition engines the carbon is so fine that it is almost impossible to remove it even by the finest filters; whereas in more normal cases and with petrol engines it is easily within the scope of filters to remove it. There is also carbon which arises from deposits which gradually form in hot positions, such as heads of pistons, and flakes off, being broken up by the motion of the parts.

(*e*) Fibres and cotton waste arising from wiping and polishing rags.

It will be generally admitted that the material under all these headings is harmful, with the possible exception of the fine carbon referred to under (*d*). In the author's opinion, however, even this fine carbon cannot possibly do any good to the engine, and in view of its tendency to agglomerate into larger particles or to settle in quiet corners, it is better to avoid it altogether.

Considerations Affecting Choice of Filter. It must be admitted that the ideal solution would be for the whole of the oil going to the bearings to be filtered completely, that is to say, that this oil would have every variety of material removed from it which was not oil. Practical difficulties in the way of achieving this are, however, so great that no scheme has yet been put forward to accomplish it. Since the rate at which oil is pumped by the average motor-car engine is in the neighbourhood of 200 gal. per hour and the rate at which a fine filter will pass oil does not exceed 1 gal. per sq. ft. even under favourable conditions, it will be seen that a filter of about 200 sq. ft. would be needed to give this performance, a filter which would be quite out of the bounds of practicability on account of size.

As a compromise there are two alternatives: either the whole of the oil may be filtered but not very finely, so that the worst of the particles are removed but still a large number of fine particles will remain in suspension; or a proportion of the oil may be filtered completely through a fine filter. At first glance it is not obvious which of the solutions offers the better advantage in practice, but in the author's opinion the bypass system, filtering part of the oil completely, would be the correct one.

Size and Effect of Particles. Taking first the case of a comparatively coarse filter dealing with the whole of the oil, it is often assumed that

the larger material, which such filters successfully remove, is that which would do the most damage. But, in fact, a very good case could be made to show that the medium-sized abrasive particles which would pass such a filter are the most active in producing abrasion. These particles are small enough to be carried along with the oil and to enter actually into the oil film, and will certainly produce a very considerable grinding and lapping effect in virtue of their number and uniformity. This will result in bearings becoming slack much more rapidly than would be the case from scratches produced by large particles, even assuming that the large particles could get into the very narrow clearances which are allowed. Admittedly, with a bypass filter, during the first minutes of its life most of the oil would not as yet have passed through the filter and, accordingly, would contain its share of whatever abrasive matter there might be. However, when the filter commences work with a new engine the oil will also be new, and the particles which will ultimately work loose and tend to cause trouble will not all be present. Consequently, the oil reaching the bearings at the commencement of the filter's work should be in a moderately good state.

Bypass Filtration. It is contended by opponents of bypass filtration —and the theoretical correctness of the contention must be admitted— that so long as only a portion of the oil is being filtered it is possible that any given particle will never be removed from the oil, as it may always happen to be present in that part of the oil which escapes the filter, and consequently can continue to do damage. A consideration of the facts, however, soon shows that such an argument, although theoretically impregnable, practically has no weight whatever. If the question is considered mathematically and it is assumed, for example, that the rate of filtration is such that all the oil in the sump would be passed in a quarter of an hour, then the chances of any particular particle of impurity remaining in the oil would be approximately:—

1,353 in 10,000	after	½ hour.
183 in 10,000	,,	1 ,,
25 in 10,000	,,	1½ hours
1 in 20,000	,,	2½ ,,
1 in 10,000,000	,,	4 ,,
1 in 500,000,000	,,	5 ,,

In view of the foregoing, the advantage of bypass filtration must be admitted since the degree of purity of the oil in the sump which is achieved by fine bypass filtration within about an hour, is superior to that which can ever be achieved by full-flow coarse filtration, or by centrifugal action.

Types of Filters. Filters of many different types have been proposed of which the following is a brief list:—

Filters employing metal gauze.
Filters employing felt.
Filters employing cloth.
Edge filters employing metal.
Edge filters employing paper, which may be cleaned.
Edge filters employing paper, of the composite type, which are renewable.
Edge filters working in conjunction with a filter bed.
Filters employing both fine and coarse filtration in a single unit.

Little need be said about gauze filters. With this type of filter the size of particles removed is limited by the size of the holes in the gauze, the action being strictly a mechanical straining-out. The finest gauze will permit particles of 2/1,000 or 3/1,000 inch to pass, and in general, coarser gauzes than these have to be employed to avoid choking, so that with gauze filters particles up to 10/1,000 inch in dimensions at least may be expected.

The next step upwards in the scale of fineness is the felt filter. This type does not rely purely on the mechanical sieving action of gauze, but in view of its greater thickness it can separate out particles which are smaller than the average size of the spaces between the constituent fibres making up the felt. Such filters have achieved a very wide sphere of usefulness, but considerable improvement is possible beyond their capabilities.

Cloth filters, which may be finer than felt, are sometimes employed. They are capable of removing quite fine particles, down to about 1/10,000 inch, but have proved rather awkward to accommodate in the engine.

Edge filters constructed of metal can be finer than felt filters in their action, and have the advantage of mechanical strength and stability much greater than that possessed by gauze filters. They are capable of a fineness of filtration about equal to that of cloth, but are unable to remove the finest suspended carbon.

Edge filters employing paper easily remove even finely divided carbon and produce a perfectly clear, transparent oil no matter what the nature of the impurities in the original oil may be. Although the output per square foot of filter surface is rather low, the fact that they may be easily, quickly, and repeatedly cleaned counterbalances this, and they are an eminently practical solution of the problem of filtering oil completely.

Where filtration is carried out separately on oil which has been

removed from the engine, metal edge-filters using filter beds may be employed. In general their manipulation is not convenient in conjunction with an engine actually working.

Edge filters of the composite type employing paper are also available. These generally consist of two disks of paper which are united at the edges and have a central drainage hole which is kept open by two metal washers placed concentrically round it (Fig. 1). These filters

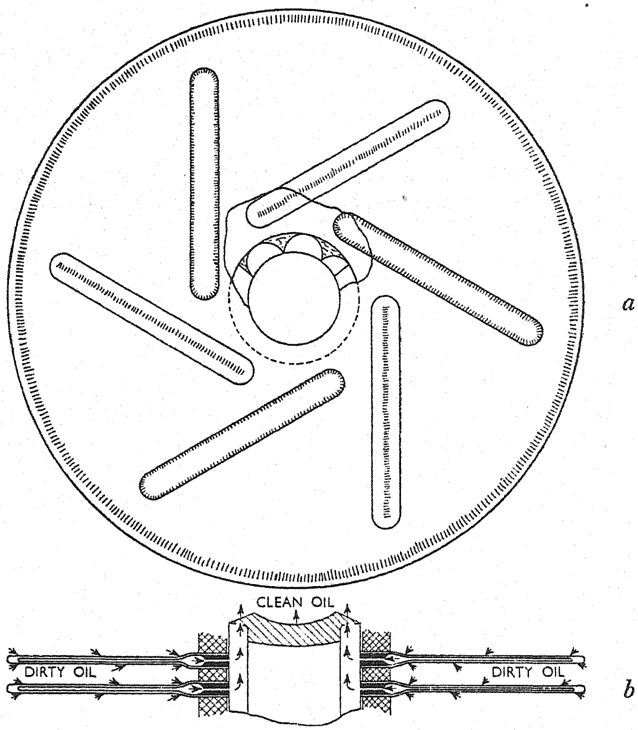


Fig. 1. Composite Edge Filter

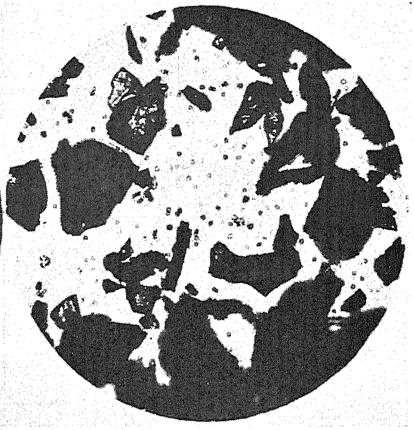
cannot be cleaned, but have to be renewed after a certain time. They are, however, comparatively rapid in their action and a very large filtering surface can be accommodated in a small volume.

In addition to the foregoing, a dual filter is possible which combines the advantages of full-flow filtration with those of bypass fine filtration. A description of the last two types of filters is given later in this paper.

Comparative Efficiency of Different Types of Filter. In reviewing the

× 125 dia.

Fig. 2. Metal Grindings after Passing through 120-mesh Sieve



× 125 dia.

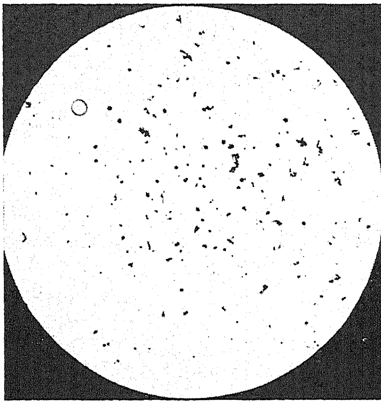
Fig. 3. Metal Grindings after Passing through Felt-Type Filter

× 125 dia.

Fig. 4. Metal Grindings after Passing through Felt-Type Filter more Tightly Adjusted

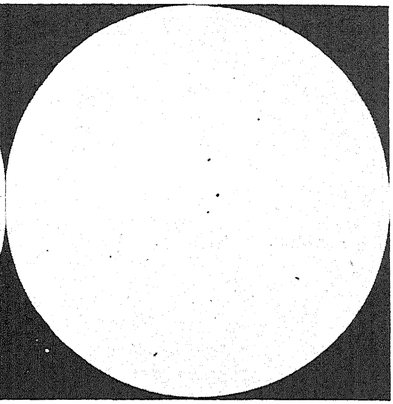
× 125 dia.

Fig. 5. Metal Grindings after Passing through a Commercial Felt-Type Filter



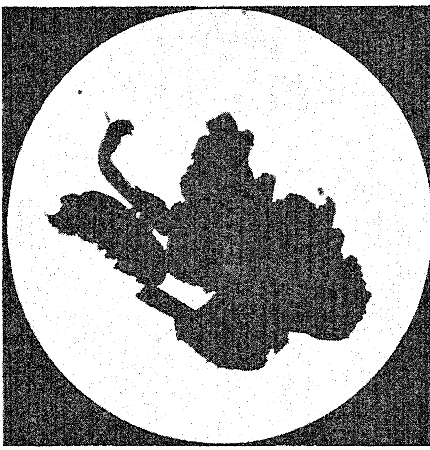
× 5 dia.

Fig. 6. Metal Grindings in Filtrate from Felt-Type Filter



× 5 dia.

Fig. 7. Metal Grindings in Filtrate from "Metafilter"



× 125 dia.

Fig. 8. Largest Individual Particle in Filtrate from Felt-Type Filter

× 125 dia.

Fig. 9. Largest Particle Passed through "Metafilter"

service given by filters of the different types some test filtrations were carried out, the results of which proved unexpectedly interesting. Some of the grindings which accumulate round an ordinary grinding wheel were taken and sifted through a sieve of 120 meshes to the inch. The material which passed through was, accordingly, very like what might be expected in engine wear as it contained iron, steel, brass, and siliceous matter, and the largest particles were not more than 0·003 inch long (Fig. 2). The sifted grindings were suspended at a concentration of 1 per cent in new, clean lubricating oil which was heated to 80 deg. C. (roughly the working temperature in a typical sump), and then passed through filters of different types. It was not necessary to repeat the gauze filtration, as all the particles had previously come through the gauze. Fig. 3 is a photomicrograph of the particles found in the filtrate from a felt filter, magnified 125 diameters. As it was thought that some of these particles might have passed through in consequence of the end joints on the filter not being fully tight, another trial was made after further tightening down with approximately the same result (Fig. 4). The experiment was also repeated with a commercial model of a filter of this sort, with the result shown in Fig. 5. It is thus clear that filters of the felt type are capable of allowing particles 0·003 inch or even larger in dimensions to pass.

A comparison was made between a metal edge-filter of standard type with 0·003 inch separation between the rings, and photomicrographs at 5 diameters magnification of the filtrates are shown in Figs. 6 and 7. It will be seen that many more and considerably larger particles came through the felt filter than through the edge filter, and Figs. 8 and 9 show the largest particles which could be found in the filtrates of the felt filter and the edge filter respectively. It will be seen that a particle passing the felt filter is many times larger than that passing the edge filter. Edge filters employing paper both of the cleanable and composite type removed all traces of suspended matter from oil of this type.

"*Pocket*" *Filters*. A description of edge filters of the composite renewable type may not be out of place, as these filters have considerable possibilities and have only recently become available. These filters are all built up of pockets which consist of two disks of a special filtering paper attached by crimping at the edges (Fig. 1*a*) and supported round the border of the central hole by two washers which are held slightly apart by projections on the inside face of one of them. Flutes are raised and depressed from the general surface of the pockets. These serve to prevent pockets which are mounted closely adjacent to each other from coming into actual contact over their whole surface. The

course of filtration may be understood from Fig. 1*b*. The pockets are mounted on a drainage pipe and enclosed in a pressure container. The oil penetrates the surface of the paper and gradually percolates to the centre of the disk, leaving the dirt deposit on the outside. The oil output may be anything between 0·02 and 0·5 gal. per sq. ft., and

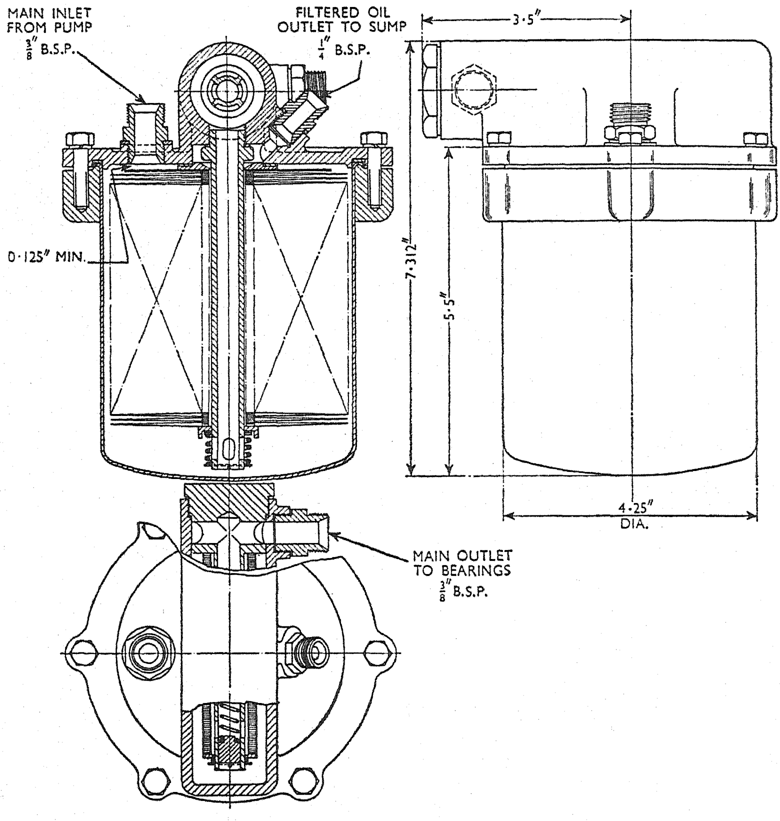


Fig. 10. Dual Filter

10 sq. ft. of filter surface can be accommodated in a container holding about a quart.

Dual Filter. A dual filter has been constructed (Fig. 10) in which the main flow of oil takes place into the body of the filter, which it enters at the top and leaves at the bottom, going upward through a

central pipe. It then encounters the coarse filter and proceeds to the bearings. A release valve is provided in the coarse filter so that should any risk of "starving" the bearings with filtered oil arise, this valve lifts and permits the oil to go forward without passing between the rings. The fine portion of the filter consists of pockets and is mounted on the outside of the central pipe. The clean oil passing the pockets and flowing up the outside of the central pipe is led to a separate outlet and returned to the crankcase. By this means the full pressure is available

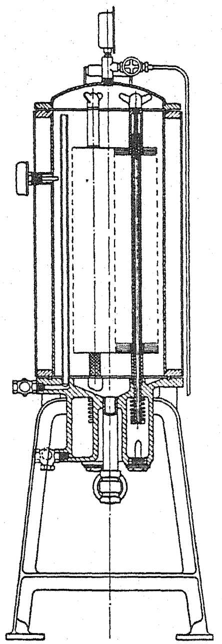


Fig. 11. Large Type of "Pocket" Filter

for the slow-working fine filter, while not very much back pressure is put on the flow of the oil to the bearings, which passes through the coarse filter. A very important advantage is also secured by this method of construction. In the ordinary fine filter a great drawback is the fact that the filter body is full of cold oil at the commencement, so that filtration must be extremely slow until this cold oil has been displaced, and as this can only be displaced through the pockets of the fine filter it will be a slow process and probably will not allow the filter to reach its full effectiveness for an hour or so after the engine has started. With the dual filter a flow of oil takes place immediately through the body of the filter as it passes through on its way to the coarse filter.

Accordingly, the temperature of the body speedily becomes the same as that of the oil in the sump, which permits the fine filter to produce its maximum output at an early point.

"Pocket" Filters of Larger Size. Filters can be built to almost any size, incorporating larger numbers of pockets. These may be employed either as batch filters or may be incorporated in the lubrication system of the engine. In a filter for batch filtration with an output of about 2 gal. per hour, the oil to be filtered is placed in the base, the small chamber at the top containing the pockets. Oil is forced by air pressure into the chamber, where it is heated by an electric jacket, and passes through the pockets. A larger filter, employing 300 pockets, with an output of about 10 gal. per hour, is shown in Fig. 11. In this case the oil is delivered by a pump and kept hot in the filter body by means of a water jacket.

CYLINDER LUBRICATION OF SMALL-BORE DIESEL ENGINES

By C. G. A. Rosen*

The life-blood of motion is the lubricating fluid, and service demands require the selection of suitable oils with as much care and analysis as for steel or other highly developed materials for specific uses. The small-bore fast-rotating Diesel motor has to maintain a supporting oil film for the surfaces of sealing and reciprocating elements in face of deposits of carbon and binder-forming materials. The large-bore Diesel engine may succeed in this quest for economical operation, but the small-bore motor has to compete under the handicap of higher unit loading, heavier service demands in the hands of unskilled attendants, and constrained oil circulating and flushing channels. This paper, therefore, deals more specifically with the influence of carbon and binder-forming tendencies on cylinder lubrication. These influences will now be discussed.

The Combustion Process. According to Boerlage and Broeze the Diesel combustion process may cause the deposition of three types of products of incomplete combustion, such as: (1) aldehydes and varnishes, produced by chilled hydroxylation; (2) soots from over-rich mixtures; and (3) the formation of products by chilled destructive combustion. These factors are influenced considerably by the characteristics of the fuel, but they are not related to the lubricant used. A fourth type of deposit may be produced by the decomposition of lubricating oils through the combustion of the fuel. These decomposition products accumulate soot from the fuel and build up sticky materials on the piston ring lands and in the piston ring grooves.

Fuel deposits vary in formation, character, and quantity according to the type of combustion system and the structure of the engine. In general, the temperatures prevailing in metal parts adjacent to the combustion chamber envelope have a direct influence upon the formation of deposits within the combustion chamber. The ring belt temperature, however, has a decided influence on the products of decomposition of the lubricating oil. In the design of turbulence chambers or contours over which burning gases must pass or impinge, great care is required to preserve sufficiently high temperatures to prevent chilling influences from altering the trend of the combustion process. A cooling surface in the combustion chamber envelope may

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be the deciding influence in producing deposits which are detrimental to the maintenance of the oil film. These influences have been discussed in the literature on the subject.

The influence of deposits from fuel becomes more marked under light loads at high altitudes, whereas under normal loads at sea level the higher temperature of the combustion chamber envelope assists to obviate or minimize these tendencies. In properly designed Diesel combustion systems, the average type of available Diesel distillate operates without detrimental deposits, provided that the ignition quality is satisfactory and the fuel is free from additions or contaminations. Experiments have been made which show the increased accumulation of combustion by-products due to the use of residual-containing fuels in loaded operations. In the ultimate analysis, gummy deposits, whether from the chemical by-products of the combustion of fuel or from the decomposition of lubricating oil, act as binders to congeal carbon and dust in the ring grooves and subsequently produce failure of the reciprocating structure.

Design of Cylinder, Piston, and Piston Ring. The design and mechanical structure of the piston and rings will influence the opportunity for carbon and lacquer deposits to form, adhere, and remain on their surfaces. Induced rubbing action of areas subject to the accumulations of deposits is advantageous. Open channels, which provide free circulation of lubricant to permit the oil to remove gummy and carbonaceous materials which seek lodgment, are beneficial. Piston design, in respect of heat flow, presents considerations of prime importance. If definite service conditions could be maintained for any given product, it is conceivable that the relation of heat flow through the ring belt to that dissipated by the skirt could be controlled so as to be within a fairly uniform range. Under such operating conditions, carbon and lacquer deposits could be controlled relatively simply by design. The variable conditions of load and temperature imposed upon engines makes this simple solution in piston design quite unpractical, if not impossible.

To illustrate the influence of ring belt temperature on deposits on the piston, the surface deposits on two pistons (Fig. 2) are compared with a common oxidation test bar. A roughly accurate picture has been obtained by the use of bar tests (Fig. 1). The apparatus consists of an aluminium bar suspended over a trough at an angle of about 3 deg. Heat is applied to the upper end until temperature equilibrium is reached. The heater is adjusted so that the temperature gradient of the bar remains constant. A 100 cu. cm. burette is filled with the oil under test, and adjusted so that the oil dripping from the burette will

strike the centre of the bar at the point of the temperature gradient, 650 deg. F. The drip is controlled at a rate of 4 to 5 cu. cm. per minute. The test lasts 6 hours under controlled atmospheric conditions. Examination of the bar indicates the type of deposit, shows the lacquers and gums, reveals hard or soft carbons, and is an index of deposition in relation to the piston ring belt section. This test is purely comparative, and evaluation of the results in terms of engine operation has been very useful. Each piston (Fig. 2) was operated in the same engine, on the same lubricant and fuel, at exactly the same brake mean effective pressure, cooling water temperature, and crankcase sump temperature. The pistons, however, provided two distinct ranges of ring

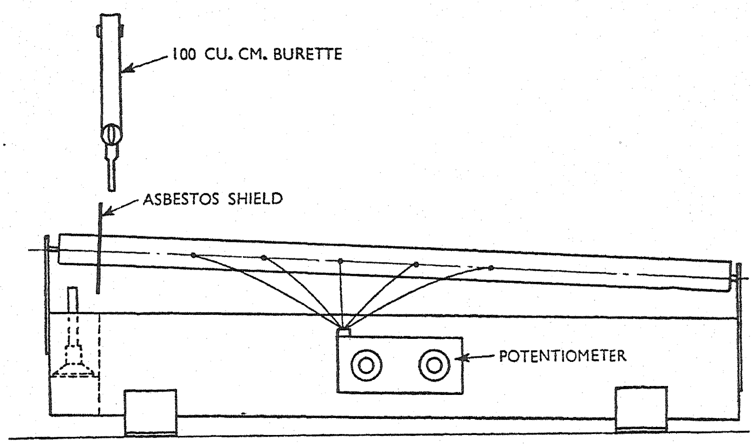


Fig. 1. Oxidation Bar Test

belt temperature. The all-aluminium piston (*a*) was designed to control the ring belt temperature within the range of heavy deposits of carbon and lacquer, as formed from a paraffin-base oil highly treated with solvent. The special piston (*c*) was so constructed as to maintain a temperature gradient above that which produced deposits on the test bar. In other words, the temperature of the entire ring belt was in the range of the clean portion of the test bar (*b*). The aluminium piston soon failed, due to aggravated ring sticking, whereas the other piston yielded higher hours of operation with complete freedom of all rings. Field data show that for comparable loads, a given design of piston will show initial sticking of the top ring when using paraffin lubricants and will affect the middle of the ring belt by ring seizure there when using naphthenic oils.

These tests illustrate some pertinent facts:—

(1) If the service and load conditions are stipulated for a given engine, a design of piston can be offered which will function satisfactorily for long hours of operation on any one given type of mineral oil.

(2) To provide for universal service over wide ranges of load under different climates, piston design must be dictated by considerations of mechanical and thermal durability, and cannot be complicated by restrictions of thermostatically controlled ring belt regions made manually adjustable to the variations of straight mineral oils.

(3) The limitations of design require that lubricants must have a minimum tendency to the formation of carbon and binder deposits which affect the proper action of the gas-sealing elements in the Diesel cycle.

Metallurgical and Mechanical Considerations. Liner, piston, and rings must be of such a nature as to enable the lubricant to establish and support the necessary oil film. For the lubricant to adhere to the liner and ring faces, graphitic carbon appears to require its share of surface. Nodular or pin-point graphite structures cause difficulties in promoting oil spreading and film adhesion. The chemical composition of the metal influences corrosion tendencies from combustion gases. Machining and finishing operations play a part in assisting the lubricant to establish an oil film. Cylinder and ring materials which tend to scratch or abrade in operation put a heavy tax on the load-carrying ability of the lubricant. The manner in which raw machined surfaces are transformed into bearing surfaces capable of maintaining an oil film is of considerable importance in relation to the life of the liner and rings. Running-in methods, even over short periods of time, endow reciprocating surfaces with a wear resistance that continues its influence after thousands of hours.

Surface Conditioning. Material improvements have been accomplished in providing rapid and safe running-in by chemical treatment of the cylinder and ring faces to endow the reciprocating surfaces with extreme-pressure characteristics. By this process piston rings, which normally require some 100 hours to obtain full surface conditioning and full bearing face, can be adequately run-in in 3 hours. Quick running-in to full surfacing is conducive to low ultimate cylinder and ring wear. The quick and effective sealing of blow-by gases to prevent them from entering the crankcase cannot be overestimated.

Ring Sticking. Certain lubricants form a lacquer film on the top surface of the piston ring which may be of insufficient thickness to

retard the floating movement of the piston ring seriously under normal operation. When, however, the engine is shut down, this lacquer shows a potent influence. An aluminium piston cools very quickly when the engine is stopped. When this cooling begins, the vertical contraction of the ring groove, which amounts to less than 0·001 inch, will tend to seize the lacquer film and clamp the ring in its groove. This clamping may not necessarily take place over the entire ring surface, but may occur only in limited regions. However, even after the ring has been clamped, the radial contraction of the piston will continue for some time, and this contraction may amount to as much as 0·020 inch in a $5\frac{3}{4}$-inch bore aluminium piston. This reduction in diameter will withdraw the clamped ring or the clamped portion of a ring from its adjacent cylinder surface, and thus provide a gap between the ring and the liner. Satisfactory continuance of operation depends upon how quickly the ring is unclamped upon subsequent starting of the engine. If channels are available for blow-by gases to sweep across the face of the ring, the ring will soon be seized fast and will ultimately collapse in the groove. This situation becomes dangerous with lubricants which tend to form hard, tenacious lacquers in the prevailing temperature range of the ring belt of the piston. High blow-by and early failure may result for apparently very little reason.

Certain lubricants produce hard granular carbon, carbon formation usually beginning in the back corners of the ring groove. If the carbon is soft and not adherent, the floating action of the ring under the alternate thrust movement of the piston will continue to break the carbon off the back of the groove, thus permitting a pumping action which flushes oil through the ring grooves. If, however, the carbon is hard, dense, and tenacious, the ring will eventually be constrained and finally clamped on the inner edge of the ring, resulting in ring sticking.

An intermediate case may well result. The character of the carbon may be such as to provide relative freedom of the ring for a considerable period, and yet may pack the back of the groove so tightly with carbon as to block the floating action of the ring. In this case there is danger of rupturing the oil film and causing failure by cylinder and ring abrasion. Sometimes soft porous materials may fill the channel openings, but may not materially hamper oil control because of the spongy absorptive quality of the material. The most serious deposits affecting oil control are those which build up like tiles, and block off the return channels, preventing the movement of oil to the back of the oil ring groove.

Position of the Oil Rings. In Diesel pistons, two oil rings are required to ensure economical oil consumption under long service. An oil

ring on the piston skirt below the piston pin is more effective at lower wall pressures than an oil ring would be at the enlarged clearances above the piston pin. However, a suitable vent must be provided for the escape of gases and other materials to the inside of the piston immediately below the bottom compression ring. This position provides an ideal position for an oil ring for venting purposes. A far cleaner piston is obtained if the bottom oil ring is also placed above the piston pin, as this eliminates the sludge trap surrounding the piston pin region. Furthermore, the improved circulation and flushing action obtained with oil rings above the piston pin are advantageous in obtaining minimum deposits and lower wear of reciprocating parts.

Ring Sticking Phenomena. To further the study of deposits of carbon and binders, a test unit was designed which permitted of rapid inspection of cylinder, piston, and rings (Fig. 3). Units of this design were operated at the continuous loads that produced ring sticking in the field in the shortest period of time. As ring sticking is developed by the accumulation of gummy deposits with respect to time, "ring-sticking hours" is assumed as the standard of measure. Operation of the test engines was continued until the blow-by of the gases from the crankcase reached values of 85 cu. ft. per hr. Normal operation is characterized by a blow-by of 12 to 20 cu. ft. per hr. in the single-cylinder unit. Tests were stopped at the high blow-by point, as this indicated serious ring sticking and as it was desired further to conserve all evidence of deposits.

A study of a number of lubricants, generally considered suitable for Diesel engines, in the single-cylinder test engine revealed that ring sticking would occur as early as 100 hours of operation on one type of lubricant, or as late as 3,000 hours on another type. In all these tests the temperature of the combustion chamber envelope was kept constant. This finding suggested that lubricating oils in themselves influenced ring sticking. Further analysis indicated that the ring-sticking hours could be tabulated in three major divisions when the lubricants were related to methods of treatment and refining (Fig. 7).

In the present state of flux in the manufacture of lubricating oils, it would be unwise to draw hard and fast lines concerning these findings. It is apparent, however, that ring-sticking hours exhibited a relationship in inverse ratio to the degree of treatment of the oil stock. Drastic treatment developed low ring-sticking hours. Minimum chemical change during finishing was conducive to long hours of operation before ring sticking developed failure. It was also evident that the source of the base stock had a significant influence on ring-sticking hours. Naphthenic base oils showed the longest ring-sticking hours in

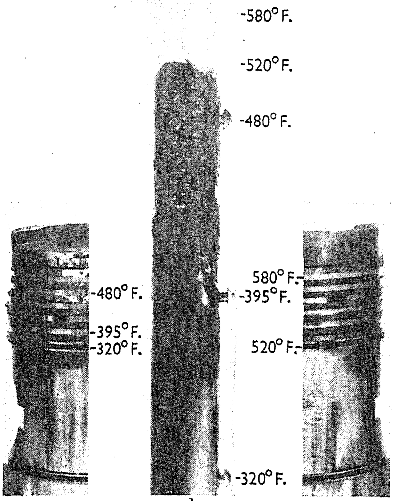


Fig. 2. Comparative Deposits on Pistons Operating in Different Ring Belt Temperature Gradients as Compared with an Oxidation Test Bar.

(*a*) Aluminium piston. (*b*) Test bar.
(*c*) Special piston.

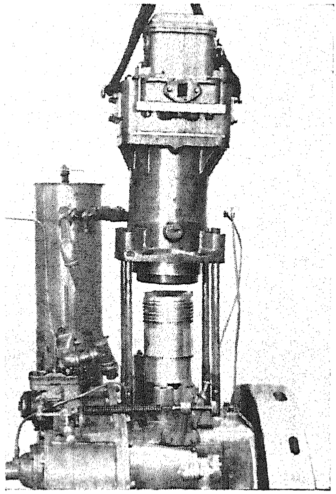
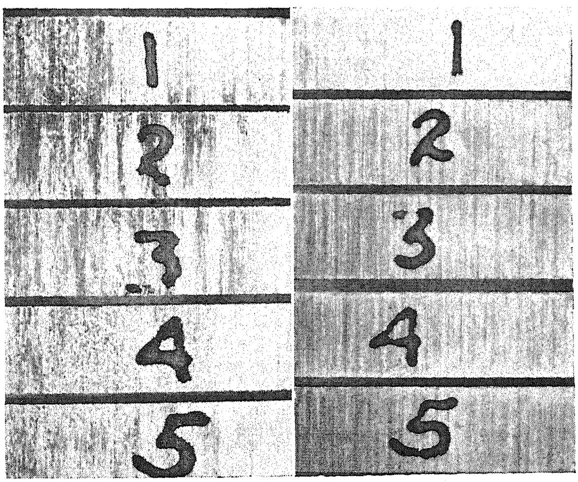
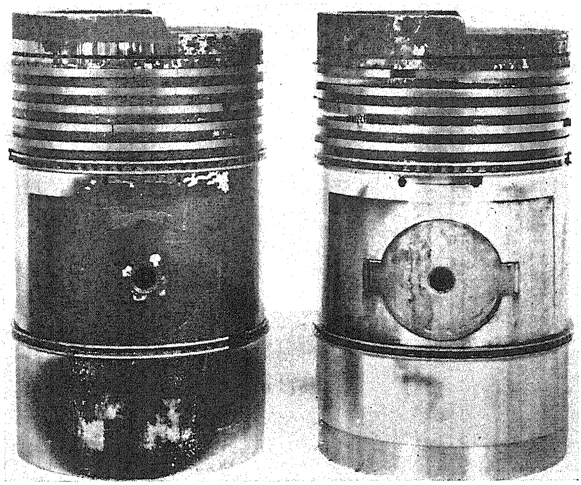


Fig. 3. "Caterpillar" Type Single-Cylinder Diesel Test Engine



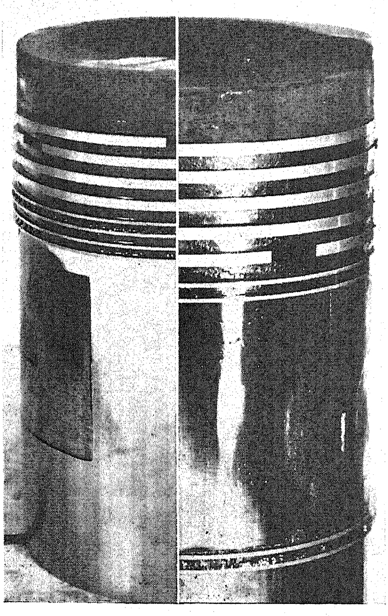
a *b*

Fig. 4. Influence of an Addition for Improving Film Strength



a *b*

Fig. 5. Improvement in Resistance to Ring Sticking due to Compounding



a *b*

Fig. 6. Influence of Compounds on Sludge Depression

each division. Results obtained at elevated ring belt temperatures showed partiality toward the more stable lubricants of high paraffinicity. Such temperatures, however, are not maintainable under broad-range service operation.

This situation leads to the conclusion that the development of lubricants pre-eminently suited to cope with Diesel engine conditions might offer greater alleviation to the ring-sticking problem than special piston ring-belt structures. Indeed, test results and operating data show that suitable base stocks of desirable treatment and finish which contain additions to deal with special problems, give superior operating results. Fig. 4 illustrates the conditioning received by the piston rings in comparable running-tests. The rings on the left (Fig. 4*a*) operated on a straight mineral oil somewhat lacking, because of treatment, in sufficient film strength to permit of full running-in without scratching. This lubricant was not capable of maintaining a film during the transition

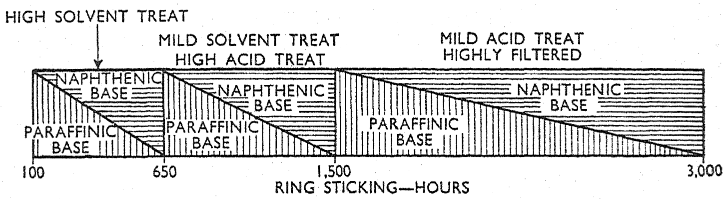


Fig. 7. Schematic Diagram of Ring-Sticking Hours in Relation to Base Stocks and Treatment

period of running-in. Fig. 5*b* shows rings which were run-in with a lubricant consisting of the same base stock as was used in the left-hand piston, but to which was added an ingredient improving film strength. No surface abrasions on the rings are evident.

A striking example of the improvement in "ring-sticking hours" by the addition of compounds to straight mineral oils is illustrated in Fig. 5. Piston (*a*) ran for less than 1,000 hours, when using a Western base straight mineral oil treated with sulphur dioxide. To this base stock was added a very small percentage of material to develop resistance to ring sticking. Piston (*b*) has run a similar period using the compounded oil; the rings are entirely free, no deposits are attached to the piston thrust surfaces, the oil rings are clean and open, and the piston pin relief still shows the machined surface. With piston (*a*) the top ring is half stuck; the second, third, and fourth rings are wholly stuck; the oil ring grooves are badly sludged; and the piston pin relief and thrust surfaces are coated with heavy, gummy deposits. Fig. 5*b* illustrates a typical case where ring-sticking hours are extended to a

point where overhaul will be occasioned by failures other than ring sticking.

The use of compounded oils also reduces cylinder liner wear. With good lubricants tested in the single-cylinder test engine, it is usual to expect a maximum rate of cylinder wear at the top of the ring travel of 0·0025–0·004 inch per 1,000 hours, after 1,000 hours operation. Some mineral oils have reached rates of 0·009–0·011 inch per 1,000 hours, whereas some of the new alloyed lubricants have recorded as low wear as 0·0003–0·0005 inch per 1,000 hours at the 1,000-hour period.

The use of additions effectively depresses sludge formation. In Fig. 6 two pistons are shown, each of which had had the same naphthenic base stock as a lubricant. The sludge deposits at 1,000 hours of the "undoped" oil are evident (Fig. 6*b*). Fig. 6*a* represents a piston

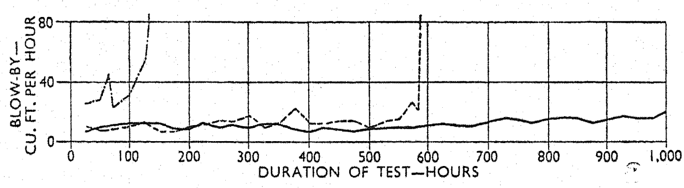


Fig. 8. "Blow-by" Shown by Three Distinct Types of Lubricating Oils

after 1,500 hours of operation with the doped oil of the same base stock and shows the absence of deposits.

Perfectly free-floating piston rings not only reduce wear but provide the piston-sealing properties which are so essential to good Diesel-engine operation. A heavy type of deposit in the piston ring-belt and on the piston skirt may be conducive to high sealing value against blow-by gases for short duration, whereas free-floating rings will permit of a certain amount of blow-by through more open areas in the ring grooves, but the consistency of blow-by never can be preserved with the high-gumming oils, whereas the non-gumming oils will maintain a very uniform blow-by rate. For example, three oils were compared (Fig. 8) for blow-by in ring sticking endurance runs. The top curve is from a lubricant yielding high piston deposits. The middle blow-by curve is illustrative of a test conducted on piston (*a*) of Fig. 5. The lowest curve is for the piston (*a*) of Fig. 6. These curves and photographs tell their own related stories. The sticking and unsticking of rings are indicated by peaks and valleys on the blow-by curve. Free-floating piston rings preserve a uniform rate of blow-by and smooth out the irregularities in the blow-by curve. Generally

speaking, the character of the blow-by curve is of greater significance than the actual blow-by rate.

Where rings are lubricated effectively and are absolutely free-floating there is a tendency to produce a stable and efficient oil consumption. By an efficient oil consumption is meant sufficient lubricant to resist wear in such regions as the top of the piston ring travel. When using straight mineral oils, which tend to produce dangerous gummy deposits, lubricating oil consumption is usually curtailed to a minimum. The fear of the formation of deleterious products of decomposition places restrictions on the amount of lubricant permitted to reach the upper portions of the cylinder surface. With lubricants which do not give dangerous deposits, there is less need to restrict the supply of lubricant to all cylinder surfaces. The use of compounded lubricants greatly facilitates copious lubrication.

In conclusion, it may be said that compounded lubricants have produced outstanding results, both in the laboratory and in the field. They are capable of use in practically all types of service, and maintain consistency of performance without dependence upon closely controlled temperature gradients. The present special Diesel engine lubricants may not have attained finality, but they point the way toward improvements in lubricants to qualify for exacting service conditions. Their value in promoting superior field performance and lowering maintenance costs already has justified their existence.

LUBRICATION AND LUBRICANTS AS APPLIED TO LOCOMOTIVE RECIPROCATING STEAM ENGINES

By W. A. Stanier, M.I.Mech.E. (*Vice-President*).*

Cylinder Lubrication. The conditions existing between the piston rings and the cylinder walls of the steam locomotive are not very well understood and they depend in some degree on the state of wear of the rings, etc. The temperature in a saturated steam cylinder may be of the order of 300 deg. F. and in a superheated steam cylinder as high as 600 deg. F. At such temperatures the oil has lost much of its original viscosity.

The methods of applying the oil vary amongst the different railways. In most cases a spray of oil is delivered through an atomizer along with the steam; in addition direct feeds to the top and bottom of the cylinder midway in the stroke are sometimes employed. With the spray method a thin film of oil may be supposed to be deposited on the walls wherever steam comes in contact with them and thus lubricant is supplied everywhere it is needed but in such small quantity that viscous film conditions can hardly be attained. The oil from a top feed to the cylinder barrel tends to flow round the walls and to some extent that from a bottom feed is pressed upward round the walls. Much of the oil is simply moved forward or backward by the piston rings, but the remainder clings to the walls and to the contact surfaces of the rings, thus effecting lubrication. Viscous films may be formed in places, but even here "boundary film" or at most "greasy film" is probably the chief means of lubrication.

In saturated steam cylinders, mineral oil alone is used and for superheated steam cylinders mineral oil with 5 per cent fatty oil is employed. This difference has resulted from practical experience, but it is important that for superheated steam the amount of fatty oil should be as low as is consistent with efficiency. Too large an amount of fatty oil results in very serious corrosion, especially when water or wet steam is present. At the temperature of superheated steam the oil becomes much less viscous and the fatty oil is partly decomposed, the decomposition products helping in the formation of stable and resistant boundary films.

There is no doubt that in the lubrication of cylinders some part is played by the graphite in the cast iron. From time to time experiments have been made with oil containing graphite, but the results

* London, Midland and Scottish Railway Company.

have not generally warranted the expense of using colloidal graphite and blocked feeds occur when graphite of coarser grades is mixed with the oil.

Oil consumption is fixed either on a time or mileage basis, depending on the system employed, and is not altered when cylinders or piston rings become worn.

The most practical finish for both the cylinder barrels and the piston rings is given by a finishing cut with a feed of 0·015 inch per revolution at an approximate speed of 250 ft. per min. A ground finish has not been found superior to this method.

Cylinder wear varies amongst the different classes of engines, the wear in an express passenger engine being approximately $\frac{3}{64}$ inch in 145,000 miles (0·00033 inch per 1,000 miles).

At one time troubles were caused by carbon which was found adhering to the piston heads, cylinder ports, and steam chests. It was found that this was not due to the lubrication used, as analysis of the deposit showed that a large proportion was foreign matter which had been drawn into the cylinders from the smokebox through the blast pipe. An investigation proved that this carbonization occurred when the engine was coasting, the vacuum produced in the cylinders being such that hot gases were drawn into the cylinders from the smokebox, thus burning away the oil film on the working parts and leaving behind carbon deposit and foreign matter from the smokebox.

The trouble of carbonization practically disappeared with the introduction of atomization of the oil, and also when steps had been taken to reduce the vacuum created when coasting by the provision of suitable steam chest air valves and by admitting a "breath" of steam to the cylinders when coasting with the valve gear at about 45 per cent cut-off.

The connexion between the chemical and physical properties of the lubricants and service performance is not well established. Oils of high viscosity are specified in order that the viscosity shall not be too low at the high temperatures attained in the cylinder. Some railways specify a range of viscosity-temperature figures more with the view of controlling the general quality of the oil than of ensuring a definite viscosity at the temperature of use. It is not usual to test steam cylinder oil with regard to its tendency to oxidize as is done regularly for oil for internal combustion engines as the oil is not exposed to so oxidizing an atmosphere. However, deposits taken from positions where the oil can remain undisturbed, i.e. the ports, sometimes show definite gumming. The analysis of such deposits indicates increased asphalt content due to oxidation of the oil, but this, generally, is not very serious and there is no clear evidence that, of the oils used, any are worse in this respect than the others. It is considered that the

information afforded by the usual laboratory tests are not applicable to the conditions existing in the steam cylinder, and further, any trouble experienced hardly warrants the devising of a new test to simulate working conditions. Coking tests are not made since the standard coking tests for oils for internal combustion engines yield results which would hardly be applicable to the lubrication of cylinders of steam locomotives. It should be remembered that the deposit in locomotive cylinders is largely derived from sources other than the oil. The pour point is always specified both to control the source of the oil and to ensure adequate feed, but it is not of significance in the actual lubrication. Absence of moisture is usually specified, but the minute amounts sometimes found can have no effect on the lubricating properties of the oil in the cylinders. Flash point is specified both to control the source of the oil and guard against the introduction of oils of low viscosity. Volatility is sometimes specified for oil for saturated steam cylinders as a control of the more viscid nature of the oil left after evaporation of the more volatile portions. As a control of the source of the oil specific gravity is sometimes laid down. Hard asphaltic matter is usually prohibited or limited by specification as it contributes to deposit in the cylinders.

The oil used for saturated steam cylinders is a mineral oil supplied to specifications indicated above. The oil for superheated steam cylinders consists of a superior mineral oil with about 5 per cent of fatty oil.

Lubricant Supply. The mechanical lubricator consists of a reservoir containing a series of small double-acting pumps giving a continuous delivery of oil. A reciprocating motion is given to the plungers of the pump by eccentrics on a shaft which is actuated by a ratchet wheel driven from some moving part of the engine which will provide a constant drive; the valve quadrant supplies such a condition.

On one type of lubricator each pump delivers 0·01555 cu. in. of oil per revolution of the lubricator shaft, the driving gear being so designed that the supply at the point of delivery is 2 oz. per 100 miles. By means of a non-slip thimble any one pump unit can be altered to deliver $3\frac{1}{2}$ oz. per 100 miles; in addition, when the engine is being run-in these amounts can be increased by one-third by means of an adjustment on the driving gear.

The point of delivery of the oil is either in the steam pipe or direct into the steam chest and also, in certain cases, to the top and bottom of the cylinder barrel. In a typical case the oil supply to the steam chest is fed through an atomizer into an annular space round the periphery of each of the piston valve liners. As it is necessary for the steam supply to the atomizer to be shut off when the engine is stationary

to prevent pressure being built up in the cylinders, this atomizer is coupled to the cylinder cock gear, so that when the cocks are open the steam to the atomizer is shut off. The feeds to the cylinder barrel are taken from the mechanical lubricator direct to back-pressure valves, loaded to open at 40 lb. per sq. in. and situated as near to the point of application as possible, to ensure that an immediate supply of oil is provided on restarting.

The hydrostatic lubricator was at one time almost universal for locomotive purposes, but although it has been to a large extent superseded by the mechanical lubricator, certain systems have been improved to meet the modern conditions of high boiler pressure, etc. When oil is supplied to a point at initial boiler pressure the extra pressure necessary to overcome that of the boiler is provided by a column of water maintained by condensed steam. On certain forms, the condensing chamber is contained in the lubricator itself, while in another a condensing coil is still used. The oil supply, which is controlled by needle valves, can be regulated as its passage is visible through sight glasses. The oil supply mixes with steam at initial boiler pressure and is carried forward to the point of delivery in an atomized condition. The point of delivery is either into the steam pipe or steam chest, reliance being placed on the flow of steam into the cylinder to carry superfluous oil with it.

One railway has designed a hydrostatic system for both saturated and superheated steam engines in which the condensation takes place in a coil fitted on the inside of the cab roof. The oil passes to a combining valve, which is governed by the movement of the regulator handle, where in a mixing chamber it is met by the steam and carried to the point of delivery in the steam pipe, being admitted through a spray stud. The combining valve is so designed that a small quantity of steam and oil is admitted to the cylinders when the engine is coasting with the regulator closed. Each feed can be adjusted separately and a control valve stops the feed through all the glasses without shutting each feed valve. The normal rate of feed is 6 to 9 drops per minute.

The main characteristics of these two methods of lubrication may be summarized as follows. With the mechanical lubricator a definite quantity of oil is delivered for every mile run, so that when the engine is travelling at high speed the rate of delivery is correspondingly increased and vice versa. Further, the amount of oil delivered is controlled by the setting of the lubricator and cannot be interfered with by the driver. In the hydrostatic system the oil is delivered on a time basis, which is quite independent of the speed of the engine; the feed is dependent on the setting by the driver and can therefore be adjusted to suit special conditions.

ENGINE LUBRICATION

By A. Taub *

Motor equipment is made for use and therefore the user is considered at every possible turn by the designing engineer and progress is being made by virtue of that consideration through the development of better materials and construction for less cost. Unfortunately an engine is not complete until its sump is filled with oil and its water jackets with a cooling medium. The motoring public has found out to its own cost that both oil and cooling medium must be suitable and must cover possible climatic variations. In the past, however, the choice was between excessive oil consumption and inability to start. Some rather poor engine oils incorporated both evils. To-day the user has more choice. Oils are available that are thin enough to give good starting when cold and sufficiently viscous under the higher temperature to give adequate protection from wear and a reasonable oil mileage. The user wants better performance, starting, and life, and his designer wants him to have it but lubricants may spoil the future. Therefore the various institutions interested in petroleum products should sanction an approved grading that will identify the cold fluidity and the viscosity ratio between 70 and 200 deg. This would permit manufacturers of motor equipment to specify the grade of oils recommended for various conditions.

Cylinder Bore Wear. Cylinder bore wear is more prevalent in Europe than in the United States. Whenever the mileage on bore life obtained in the United States is given, results of "soft" tests in Europe are quoted to show that here and there it is not *so* bad. Experience in both continents indicates that bore wear is much more severe in Europe than in America. Six cars operated by the Atlantic Refining Company two years ago showed an average bore wear of 140,000 miles per 1/1,000 inch of wear. The compression had increased slightly by the end of the test and the rings had opened at their gaps an average of 0·012 inch These figures are the average of three makers of cars and are remarkable for ring life.

Experience indicates that bore wear must be met by the following considerations :—

(1) A copious oil supply to the bores as quickly as possible after the engine has been started.

* Vauxhall Motors, Luton.

(2) Adequate blow-by and oil control at the piston rings, not at the oil supply.
(3) Temperature regulation by thermostat.
(4) Adequate ventilation of the crankcase.

Piston Rings. There has been a great deal of progress in rings in the United States during the last seventeen years. From 1920 to 1927 American rings were of uniform section and uniform radial pressure and were hammered to this pressure. On the average the rings had a diametral tension of 7–9 lb., the ratio of radial thickness to diameter was 28/1, and the ring was finished with sharp corners, there being no restriction to methods of casting, except that the ring must be free from shrinkage strains. Up to 1932 rings were individually cast for longer life and higher tension, 9–11 lb. diametral tension being aimed at. The radial pressure was uniform and the radial thickness ratio was 24/1. Hammering to shape had been eliminated. Up to 1937 rings were and are individually cast and a pressure pattern cast in to eliminate uniform radial pressure. The diametral tension was 12–17 lb., the radial thickness ratio was $22\frac{1}{2}/1$, and the face was tapered. As the power output increased, so did the tension of the rings to give better general operation, specifically to give both oil control and better blow-by control.

In Europe, up to 1928, the average rings for which records are available were hammered to shape for uniform radial pressure. The diametral tension was 3–5 lb., the radial thickness ratio was $29\frac{1}{2}/1$, and the sides were honed or lapped. These conditions prevailed up to 1934. Up to 1937, rings were hammered to shape and heat-formed to give $6–8\frac{1}{2}$ lb. diametral tension. The radial thickness ratio was 28/1. Comparison of these periods of development shows a remarkable difference in ring history between here and abroad.

There were reasons for these changes in the transatlantic rings. Hammering of rings was discontinued because this peening weakened the ring locally at the hammer marks, and for durability was replaced with the individual cast ring that could be made stiffer through the control of the shape of the original casting while the uniform radial pressure then considered necessary could be easily controlled. Blow-by became a serious factor with a brake mean effective pressure above 107 lb. per sq. in. and more tension was required. Ring chatter became a factor in blow-by. In 1932 serious consideration to the ring as a spring was given. There is an intermittent pressure behind piston rings whose frequency is controlled by the speed of the engine. This frequency may cause the ring to flutter in its natural period to a point where the ends butt so hard that they break the rings pro-

gressively until there are three or four pieces about $\frac{3}{16}$ inch long at the ends of each ring. High point pressure, or a calibrated pressure pattern replaced the uniform radial pressure ring. When a ring chatters the blow-by is excessive and very destructive. The usual result of blow-by, in addition to destruction by heat and leaks, is dilution of the oil. While the oil is being loaded with acid and the rings and bores are being spoiled by the blow-by, the oil film is broken through. Instead of replenishing the film with good oil it is fed with acid-loaded oil. There is nothing so destructive to an engine as excessive blow-by. High-tension rings with a pressure pattern cast into the ring as an individual casting provide a solution for excessive blow-by.

The radial pressure now is not uniform but has a definite pattern that is calibrated to give the highest resistance to chatter and minimum blow-by through the range. The finished ring is absolutely round in the bore and although the diametral tension is relatively high there is a point of low tension in the ring which is 90 deg. from the gap. The pressure is much higher at the points and back. This pressure pattern is important and is preferably built in during the original casting because it is necessary to maintain this characteristic for the longest time. If this characteristic is built in after the ring has been machined in full or part, usually the effect "wears off".

Tapered facing is included to permit the compression ring to carry the oil control load until the oil ring is seated in. It takes 2,000 miles to wear off the taper and just as long to run the oil ring in. When the taper is kept to 0·0005 to 0·001 inch per inch the flexibility is unchanged and the tendency to blow-by is not increased.

High-tension rings do not increase bore wear as the outward pressure of the rings due to their own tension is a small percentage of the pressure of the rings on the walls. The pressure behind the rings is much the greater percentage of the total pressure. Ring tension controls ring action, but the pressure behind the ring certainly is most important from the standpoint of seal. High-tension rings are being used in engines that are relatively free from bore wear, and low-pressure rings are being used in engines that are prone to bore wear. This proves nothing, except that high-tension rings in themselves do not promote wear. There is a great deal of information lacking on piston ring design. Being able to cast a particular form is a distinct advantage, but more information is required to show what that form should be.

With proper compression rings and a large oil supply, a good oil control is required. Inconsistencies of oil mileage are well known. Oil consumption must not be confused with oil supply. A generous

oil supply is a fundamental necessity. Good oil mileage is good engineering if durability is not sacrificed for oil economy, and it need not be. Oil mileage is a sensitive barometer for slight inaccuracies. For instance, the oil consumed with bores kept round within 0·0005 inch and straight to 0·001 inch is half that consumed with bores held round to 0·0015 inch and straight to 0·002 inch.

The piston skirt taper should rightly be large at the bottom. Due to tolerance the skirt may be 0·0005 inch smaller at the bottom than near the rings, or it may be 0·001 inch larger at the bottom than near the rings. The oil consumed with reverse taper is double that with proper taper. Assuming that the allowable tolerance for diametral tension is 9 to 13 lb. the difference is a 50 per cent increase in oil used with lower tension.

Tolerance is usually allowed in the ovality of the piston for the difference between the major and minor axis and the difference in oil used between 0·004 and 0·008 inch ovality is 40 per cent in favour of the small ovality.

What happens when all the good points combine in one engine and all the bad in another accounts for the wide difference, such as variations from 300 to 2,000 miles per quart of oil, found in engines built to the same specification. Evaluation of the fundamentals is an extensive and laborious undertaking, and requires complete control of all possible dimensions in many engines, so that each value can be rightly determined. There are many more so-called minor factors that have major effects on oil consumption such as piston fit, ring fit in the grooves, sharpness of ring edges and, of course, the oil viscosity and operating temperature.

The oil ring details have a major effect. A slotted ring must have the widest possible slots, because a little blow-by will cause the slots to choke with carbon. Drilled rings fill more slowly with carbon than do slots, and the drilled ring is less likely to break in handling. The width of the scraper lands is important: 0·024 inch seems to be the minimum possible without undue wear, but a maximum of 0·033 inch, or 0·009 inch more, must be accepted if the minimum width is to be certain. Yet there is a 30 per cent difference in oil used for the same tension with these widths.

It is by knowing these things and exercising all the control possible, that this confusion can be translated into terms of oil consumption. It must be understood, however, that an oil control can be established to cure any stable condition, and give excellent mileage results, no matter how bad the condition and no matter how great the oil supply. However, since we must not over-control, all the knowledge cannot be combined in an application unless the condition that demands super-

control is stable. In other words, a thoroughly bad job can be corrected providing it is always in need of the correction.

Erosion. Wear from abrasion has not been discussed because this type of wear in England can come only from two sources—one which should not exist and one which is readily offset. They are foreign matter in the oil, and road dust. The first is due to carelessness and is uncommon, and the second, road dust, is not very serious. The wet type of air cleaner will suffice for any condition liable to be met with in Great Britain.

Wear due to erosion and corrosion has been discussed together because it is not known where one begins and the other ends. The wear due to erosion is the most prevalent type of wear except in such engines as are definitely over-cooled and kept so through misunderstanding. It is safe to say that this is not the general rule. There is nothing that may be charged to erosion that cannot be overcome by adequate oil, with piston rings to suit. High spots or blisters on the piston, due to heat, will not make dry spots on the cylinder wall if the dry spot is instantly wetted with new oil. Excessive blow-by alone will defy oil because it will blow or burn the oil away and attack pistons and rings and bores. Therefore there must be plenty of oil, and adequate oil and blow-by control. Corrosion of the bores can be mitigated by good rings and oil and in addition there must be a quick warm up, with proper ventilation, but first of all oil is required. Light engine oils must be more generally used.

Admitting that oil consumption increases as the viscosity decreases, it is still possible to operate, with more than just satisfactory oil mileage, with oil much thinner than any in general use. Evidence of under-oiling of automotive equipment in England is provided by the fact that certain transatlantic oil pumps have twice the oil capacity of certain pumps in English engines.

PROBLEMS IN THE FIELD OF INTERNAL COMBUSTION ENGINE LUBRICATION

By Professor C. Fayette Taylor *

In spite of the tremendous amount of very fine research work which has been and is being done in the field of internal combustion engine lubrication, it is still impossible to design bearings, or even to select a lubricant, without relying heavily upon field experience and rule-of-thumb methods. It is the purpose of this paper to outline the nature of the many unsolved problems in this field and to make suggestions as to lines of attack which might lead towards their solution.

Journal Bearings. In view of the well-developed theory of the journal bearing operating under steady conditions, it is surprising to consider that criteria for the design of such bearings are still uncertain, and that the designer must depend largely on previous practical experience, plus a considerable amount of guesswork, in deciding on the dimensions of the bearings in any new design. None of the various formulæ proposed for determining maximum allowable bearing loads are satisfactory, for it always seems possible to increase such "limiting" loads by proper design.

The reasons for the discrepancies between theory and practice in the case of journal bearings are not hard to find. First, theory and experiment have been confined largely to steady loads, whereas in the internal combustion engine bearing loads vary both in direction and magnitude. An investigation of the effect of load variation on the carrying capacity of journal bearings would be very useful.

Another reason why theory and practice do not agree in this field is that theory always assumes rigid bearings, while in practice, bearings and journals are never rigid. What are the deflexions in the bearings of internal combustion engines, and what is their effect on bearing carrying capacity ? Here are two important questions, the answers to which are not yet available.

Another important factor in bearing performance is the working temperature of the oil film. Very little information is available for engine bearings in actual operation, and even less on the rates of heat flow which control such temperatures. Research in this field is badly needed.

In recent years much has been done in the development of bearing

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materials. However, this has been largely on a "cut and try" basis, and very little fundamental knowledge is available on the physics and chemistry of bearing metal behaviour under actual working conditions.

Another useful investigation would be a determination of the engine conditions under which the journal bearings operate in the partial-film region, and to what extent such operation can be tolerated. Indications are that continuous operation in the partial-film region is not possible with heavily loaded, high-speed bearings, such as those used for crankshafts and crankpins, but definite information is lacking.

Pistons and Piston Rings. No satisfactory theoretical treatment for the lubrication of these bearings has been developed to correspond with the theory for rigid journal bearings. Investigation of the basic factors controlling friction and lubrication for reciprocating bearing surfaces would be of great general interest.

It seems fairly well established that pistons and piston rings operate most of the time in the partial-film region. Of interest in this connexion are some experiments made recently at the Massachusetts Institute of Technology by means of a six-cylinder automobile engine from which all valves had been removed, and the intake and exhaust manifold openings plugged (Taylor 1936). By introducing air pressure into the manifold-cylinder system, it was possible to impose a known constant pressure on the pistons and to measure the resultant friction effects by "motoring" with a cradle dynamometer. Fig. 1 shows relationships obtained between engine friction and piston load. Journal bearings operating in the complete-film region show substantially no variation in friction with load, when viscosity and speed are held constant. The influence of load shown in Fig. 1 would indicate partial-film lubrication for pistons.

Fig. 2 shows the relationship between oil viscosity and friction, as determined with the same equipment. For journal bearings operating in the complete-film region, friction tends to be directly proportional to viscosity, other factors remaining constant. For the pistons, Fig. 2 shows that friction is not directly proportional to viscosity, which is further evidence that partial-film conditions prevail.

Fig. 3 shows a typical oil consumption–speed curve for a gasolene engine. It is possible that the sharp rise in the curve at high speeds may indicate a transition from partial-film to complete-film lubrication. Whether this is so or not has never been definitely established, but it opens up interesting possibilities for investigation.

While not strictly questions of lubrication, problems of oil consumption and piston-ring sticking are so closely allied to lubrication as to require discussion. The problem of oil consumption is now fairly

well understood from a practical point of view, although little detailed knowledge is available concerning the mechanism of oil flow past pistons and piston rings. A study of the kinetics of this problem might yield interesting results. As for the sticking of piston rings, much empirical information is already available, but again it seems difficult, on the basis of present knowledge, to reduce this question to its fundamental terms. The pressing nature of this problem in many

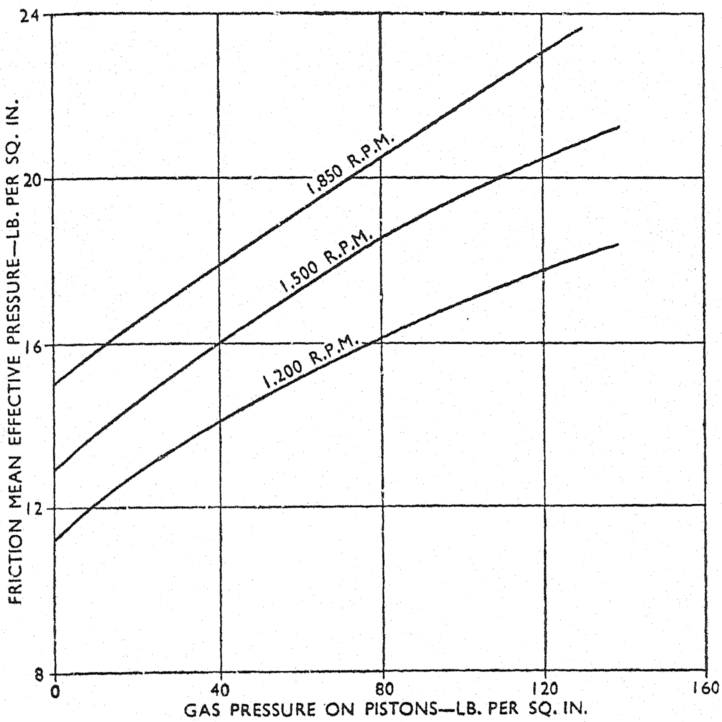

Fig. 1. Effect of Gas Pressure on Engine Friction

types of service leaves little doubt that our knowledge here will increase rapidly.

Measurement of Engine Friction. Closely allied to the question of engine lubrication is that of the measurement of engine friction (McLeod 1937, Moss 1927). Not only are methods of measuring the total engine friction unsatisfactory, but our knowledge of the friction in the various parts of the engine under actual operating conditions is exceedingly small. While accepted on account of its convenience, the

"motoring" test is admittedly a poor measure of friction under actual operating conditions. High-speed engine indicators still carry the burden of proof that their readings are sufficiently accurate to give reliable indications of friction by the difference between brake and indicated output. In multicylinder engines, moreover, it is necessary to indicate all cylinders simultaneously, so that even a thoroughly accurate indicator still leaves us with a cumbersome method.

When it comes to measurement of the friction of various parts of the engine, we are faced with a still more difficult problem, and this affords a field for investigation which should prove of considerable interest.

Oils. The optimum oil properties for journal bearings, pistons, rings and anti-friction bearings are probably quite different, and yet a

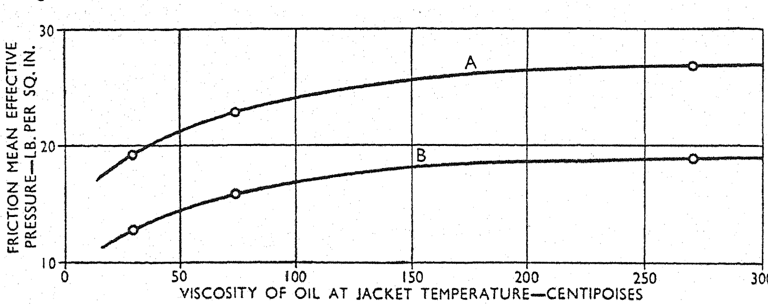

Fig. 2. Effect of Oil Viscosity on Engine Friction

A No pressure on pistons.
B Pressure of 100 lb. per sq. in. on pistons.

single oil must be chosen which will serve all these points satisfactorily. At the same time such an oil must not give trouble from ring sticking, excessive carbon formation, or corrosion, and its consumption must not be excessive. Is it any wonder that oils for internal combustion engines must be chosen on an empirical basis? For example, the choice of the best viscosity for use with a given engine under a given set of steady operating conditions involves a compromise between friction, wear, and oil consumption which, on the basis of present knowledge, cannot be made on a quantitative basis. A determination of the most economical oil viscosity for a number of typical engines, while very difficult to perform properly, might be productive of very useful results.

As loading conditions on engine bearing surfaces become more severe, more and more interest attaches to those qualities of an oil, independent of its viscosity, which affect friction, wear, and the

tendency to "seize", in the partial-film region. These have been spoken of together as the property of "oiliness", although it would seem better to restrict this term to that property of the oil which affects friction only, at a given viscosity. There is considerable evidence to show that the characteristics of an oil which effect wear and seizure are not necessarily related to "oiliness" or to each other (Beall 1937, Davis, Lincoln, and Sibley 1936, Otto, Miller, Blackwood, and Davis 1934). A considerable amount of information is available on the effect

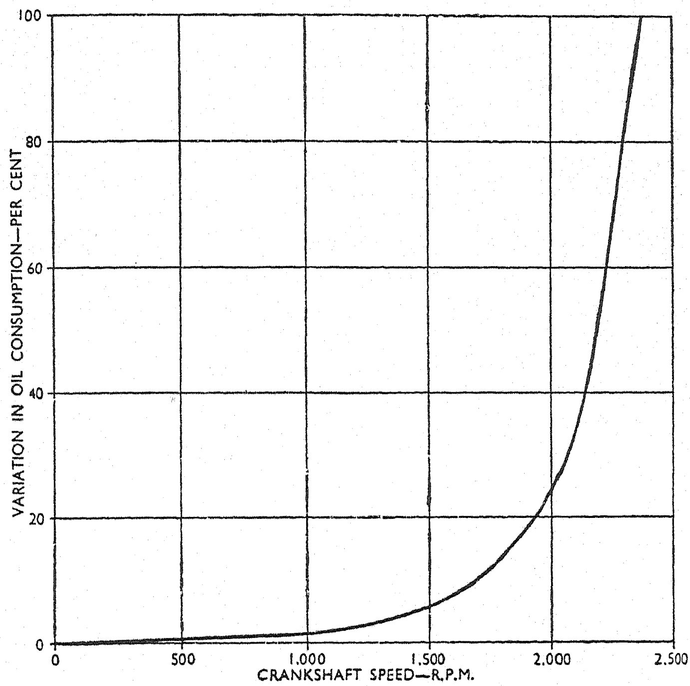

Fig. 3. Typical Variation of Oil Consumption with Speed in a Radial Aircraft Engine (Taylor 1931)

of certain oils and blending agents on these properties, but the mechanism by which these tendencies are controlled remains completely obscure.

It is encouraging to note the careful manner in which the question of the effect of oil composition on ring sticking has been attacked in many laboratories (Larson 1937). Here again, blending agents of various kinds have been found to produce beneficial effects. It is hoped that continued research will reveal the basic reasons for such action.

Conclusion. A review of the subject matter and authorship of the various papers presented at this Discussion would indicate that many of the problems mentioned herewith are being attacked with enormous energy and effectiveness. If, however, this paper serves to emphasize the need of continued research in this most interesting field, it will have served its purpose.

References

BEALL, A. L. 1937 S.A.E. Journal, vol. 40, p. 48. "Selection of Oils for High Output Aircraft Engines."

DAVIS, L.L., LINCOLN, B. H., and SIBLEY, B. E. 1936 Paper read at 16th Annual Meeting, Am. Chemical Soc. "Practical Selection of Improved Lubricants."

LARSON, C. M. 1937 Paper read at Tractor Meeting, Soc. Automotive Eng., April 1937. "Effect of Addition Agents in Lubricating Oil on Piston and Ring Performance in Gasolene and Diesel Engines."

MCLEOD, M. K. 1937 Paper read at Annual Meeting, Soc. Automotive Eng., January 1937. "The Measurement of Engine Friction."

MOSS, H. 1927 Aeronautical Research Committee, Reports and Memoranda No. 1128, 1927–8, vol. 2, p. 838. "Motoring Losses in Internal Combustion Engines."

OTTO, M., MILLER, F. L., BLACKWOOD, A. G., and DAVIS, H. G. 1934 Proc. Am. Petroleum Inst., 15th Annual Meeting, Section 3, pp. 76–89. "Improved Motor Oils."

TAYLOR, P. B. 1931 S.A.E. Journal, vol. 26, p. 531. "Increasing the Thrust Horse-power from Radial Air-cooled Engines."

TAYLOR, M. P. 1936 S.A.E. Journal, vol. 38, p. 200. "The Effect of Gas Pressure on Piston Friction."

LUBRICATION PRACTICE IN DENMARK

By T. C. Thomsen, M.I.Mech.E.*

Lubricating oils must possess various characteristics according to the service the oil is called upon to render. These characteristics will be reviewed very briefly in relation to service requirements.

Viscosity. Oils must possess a certain viscosity according to the speed, temperature, and pressure prevailing. When fluid friction can be obtained, then it is only the correct grade of viscosity that is required in order to produce fluid friction, i.e. a minimum of frictional loss.

Oiliness. When fluid friction cannot be obtained, viscosity alone does not suffice; the oil must possess a certain degree of oiliness, and this property becomes particularly important when operating conditions are difficult, i.e. the speed is low, the pressure or temperature is high, the oil feed is limited, or the formation of a complete oil film is difficult because of unsatisfactory oil distribution to the frictional surfaces, or other causes.

Stability. Oils for certain applications must withstand the action of *water* (non-emulsifying property), *air* (non-oxidizing property), *heat* (non-carbonizing property), or *cold* (low cold test).

Emulsifiability. For certain conditions of lubrication, where the oil feed is limited and where water unavoidably mixes with the oil, the oil must possess emulsifying properties, as straight mineral oils would be displaced from the frictional surfaces by the water. There are also other conditions for which emulsifying properties are desired, as in the splash lubrication of high-speed enclosed type steam engines.

Special Properties. Oils used for external lubrication get on to bright parts of machinery, the surface becoming covered by a dark coloured skin, particularly when exposed to strong light. For such conditions, pale oils are better than dark oils. The use of compounded oils containing a small percentage of animal oil or fat is desirable, since the fatty oil prevents the skin from becoming intimately attached to the surfaces, so that they can be wiped clean.

In textile mills, lubricating oil may get on to the fabric, and must be scoured out. The oil must therefore be easily removable.

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Cutting and cooling fluids, when used in circulation systems, are apt under certain conditions to become infected and cause skin diseases. A suitable soluble disinfectant should therefore be added.

Ordinary and High-Grade Lubricating Oils

During recent years, considerable advance has been made in processes for the production of lubricating oils. As a result, the oil refiner is to-day less dependent upon the character and nature of his raw material than was the case formerly. The trend of development is therefore towards producing two groups of lubricating oils, i.e. ordinary lubricating oils, and high-grade lubricating oils.

Ordinary Lubricating Oils. These oils are used for the external lubrication of all kinds of machinery where the oil is not specially exposed to the action of water, air, heat, or cold. They may either be pure mineral oils, or, where great oiliness is required, they may be mixed with a small percentage of vegetable oil or fatty acid. Where low temperatures are met, these oils may be mixed with cold test inhibitors such as "Paraflow," but they do not possess particularly good non-emulsifying, non-oxidizing, non-carbonizing, or other special properties.

High-Grade Lubricating Oils. These oils are treated by special processes so as to have special properties according to the particular class of engines or machinery involved. Where great oiliness is required, they may be mixed with a small percentage of vegetable or animal oil, or fatty acid, and where they are exposed to low temperature, they are mixed with cold test inhibitors. Under certain conditions, the admixture of colloidal graphite is desirable.

Saturated Steam Engines. The oil is exposed to wet steam and to high temperature. Conditions call for a mineral lubricating oil of high viscosity, preferably compounded with from 3–12 per cent of animal oil or fat so that the oil will emulsify with the condensed steam.

Superheated Steam Engines. As the oil is exposed to high temperature, excessive oil feed will lead to trouble owing to deposits of carbon. A highly refined mineral lubricating oil of high viscosity is required, preferably compounded with 3 per cent of animal oil in order to increase oiliness.

Exhaust Steam Conditions. When the exhaust steam is condensed and used again as boiler feed water, straight mineral cylinder oils are to be preferred both for saturated and superheated steam engines, though cylinder oils which are very slightly compounded can be used.

Blowing Engines. The oil is exposed to the oxidizing influence of

air at a moderate temperature, as the final air pressure is not high (10–30 lb. per sq. in.). A high-grade mineral oil with reasonably good resistance to oxidation is required.

Air Compressors. Excess of oil must be avoided, as the oil is exposed to air at high temperature. In multistage high-pressure air compressors (such as Diesel compressors) the oil is also exposed to the action of water (moist air) in the later stages of compression. A high-grade pale mineral oil is required, with good resistance to oxidation, and, where moist air is present, the oil should be compounded with 3 per cent of non-drying animal or vegetable oil, or up to 0·5 per cent of free fatty acid.

Air-Operated Engines and Tools. The speeds are high, and the oil is exposed to low temperature, so that a mineral oil of low viscosity and giving a low cold test, is required. When expansion is carried so far that moisture in the air is liable to produce snow, the oil must be mixed with up to 50 per cent of glycerin to keep the exhaust passages clear.

Refrigerating Machines. The oil is exposed to high temperature in the compressors, and some oil gets carried through into the refrigerator coils where it is exposed to low temperature. A high-grade, pale mineral oil with low cold test and non-oxidizing properties is required.

Gas Engines. The oil feed is preferably sparing, to avoid carbonization. Where producer gas is employed the oil may be exposed to moisture. When the gas is dry and clean, conditions call for a high grade, pale mineral oil resistant to carbonization, and preferably (but not necessarily) compounded with up to 6 per cent of non-drying fixed oil.

Petrol Engines. Conditions call for a high-grade mineral oil with good cold test, non-oxidizing, and non-carbonizing properties. In the rotary type of aeroplane engine, the oil consumption is extremely high. Castor oil produces gummy deposits in the crankcase, necessitating frequent cleaning, but it is the only oil which can be consumed in large quantities without leaving excessive carbon deposits.

An admixture of up to 10 per cent of non-drying fixed oil, or up to 0·5 per cent of free fatty acid increases oiliness and is desirable for internal lubrication, but has the disadvantage of being likely to lead to oxidation deposits in the crankcase. For this reason, automobile oils generally consist of pure mineral oil. Corrosion inhibitors are now added to automobile oils to minimize the attacks on the metallic surfaces which take place every time the engine starts up from cold and until the internal frictional surfaces and the cooling water have attained a sufficiently high temperature.

Paraffin Oil or Semi-Diesel Engines. When combustion is clean,

and no water is added to the hot bulb, a high-grade, pure mineral oil with non-oxidizing and non-carbonizing properties is required. Up to 10 per cent of non-drying fixed oil, or up to 0·5 per cent of free fatty acid may be added to increase oiliness and to ensure clean lubrication. When combustion is not clean, and, or alternatively, when water is added to the hot bulb, up to 15 per cent of non-drying fixed oil, or up to 3 per cent of free fatty acid, should be added to the oil.

Diesel Engines. As the oil is exposed to intense oxidation on the compression stroke, the lubricant should be a high-grade mineral oil resistant to oxidation and carbonization. Great oiliness is also required, so that when pistons are lubricated independently from the external lubrication, the oil may be compounded with up to 6 per cent of non-drying fixed oil.

Circulation Oiling Systems. With circulation oiling systems in which the oil is not exposed to water or high temperature, ordinary lubricating oils can be used. With circulation oiling systems for high-speed, enclosed type steam engines or steam turbines, the oil comes into contact with water, and so must withstand emulsification. When the oil simultaneously is exposed to high temperature (60 deg. C. or above), it must also resist oxidation. In the circulation oiling system for internal combustion engines of all kinds, the oil is exposed to oxidation, and, if there is leakage of water into the oil from the cooling system, to emulsification. In general, therefore, circulation oils require good non-oxidizing and non-emulsifying properties.

Splash Oiling. Some high-speed enclosed type, vertical steam engines are splash-lubricated, the crankcase containing a bath of water with a cover of oil from $\frac{1}{8}$ to $\frac{1}{4}$ inch thick. The oil is exposed to rather high temperature and emulsifies with the water. A suitable oil would be a high-grade, filtered cylinder oil, compounded with up to 6 per cent of non-drying animal oil.

Ball and Roller Bearings. Conditions call for high-grade mineral oil or pure neutral lubricating cup grease with a suitable melting point.

Locomotive and Marine Steam Engines. The oil feed is limited, and bearing pressures are high. The bearings are exposed to leakages of condensed steam from the stuffing boxes, and are often exposed to low temperature. The lubricant can be an ordinary mineral oil with good cold test, compounded with up to 25 per cent of fixed oil, usually blown vegetable oil of high viscosity, to increase oiliness and give emulsifying properties.

Some marine engines are enclosed, and the external lubrication is effected by means of a circulation oiling system employing pure mineral oils of high viscosity. The entry of water into the oil should

be prevented, and the oil in circulation should be freed from water, preferably by centrifugal treatment.

High-Speed Spindles. High-speed spindles in textile mills require ordinary pale mineral oil of low viscosity and great oiliness. If compounded with up to 10 per cent of non-drying fixed oil, oils of lower viscosity can be used, thus decreasing power consumption, and owing to the increased oiliness, lower oil consumption is also made possible.

OIL VISCOSITY IN RELATION TO CYLINDER WEAR

By C. G. Williams, M.Sc.*

There is a good deal of interest in oils of low viscosity and a definite trend towards their adoption. While the advantages of light oils in regard to cold-weather starting are undoubted, it is important that the performance of light oils should be considered in all its aspects in order that both manufacturer and operator can appreciate all that may be involved in their adoption. Such aspects include: lubricating oil consumption, "blow-by", engine friction, bearing temperatures, bearing life, and cylinder wear.

When conditions of fluid lubrication exist, viscosity is the property of the lubricant which determines the thickness of the oil film. There are reasons to believe, however, that in cylinder lubrication fluid conditions do not exist at the ends of the piston travel and it is, of course, in the region of top dead centre that maximum wear occurs. Experiments † have shown the absence of any appreciable effect due to viscosity under abrasion conditions, wear being approximately constant with cylinder wall temperatures ranging from 90 to 275 deg. C. Wear was also largely independent of the amount of dilution of the lubricant with kerosene.

The present experiments were carried out on four lubricants A, B, C, and D, derived from the same crude oil, whose viscosity-temperature characteristics are given in Fig. 1. Oil A is the "heaviest" or most viscous oil, B the next heaviest, and so on. Oil D conforms to the American S.A.E. 10W specification, and is therefore an ultra-light oil. Tests were carried out to ascertain the relative rates of wear: (1) under hot-running or abrasion conditions; (2) under intermittent operation; (3) under continuous cold-running; and also (4) to study the effect of oil viscosity on the time taken for an oil film to form on the cylinder walls after starting from cold. Single-cylinder engines of $3\frac{3}{8}$ inches bore were used for the wear experiments, the operating conditions being specified on the appropriate graphs.

The hot-running experiments were carried out on an overhead-valve air-cooled unit, and the results are indicated diagrammatically in Fig. 2. Considering the tests as a whole, there appeared to be a slight tendency for wear to be higher with the lighter oils, though some of

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* Director of Research, Institution of Automobile Engineers.

† Interim report on Cylinder Wear, Jl. Inst. Automobile Eng., 1933, June, p. 71.

this increase may be only apparent as it is almost within the limits of experimental error. Fig. 2*c* gives the results of tests on all four lubricants and indicates no appreciable change in piston ring wear over

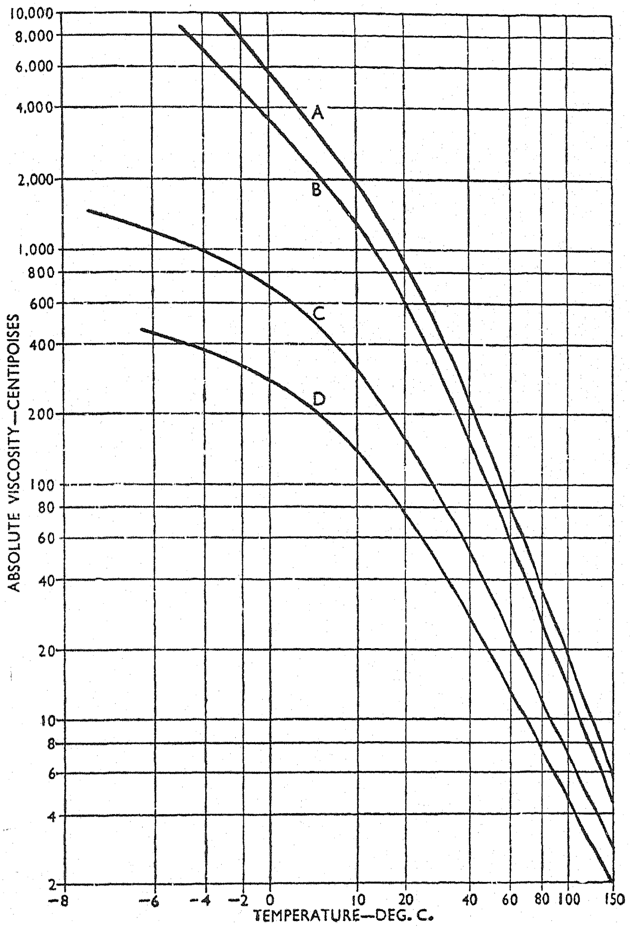


Fig. 1. Viscosity–Temperature Characteristics of the Lubricants Employed in the Tests

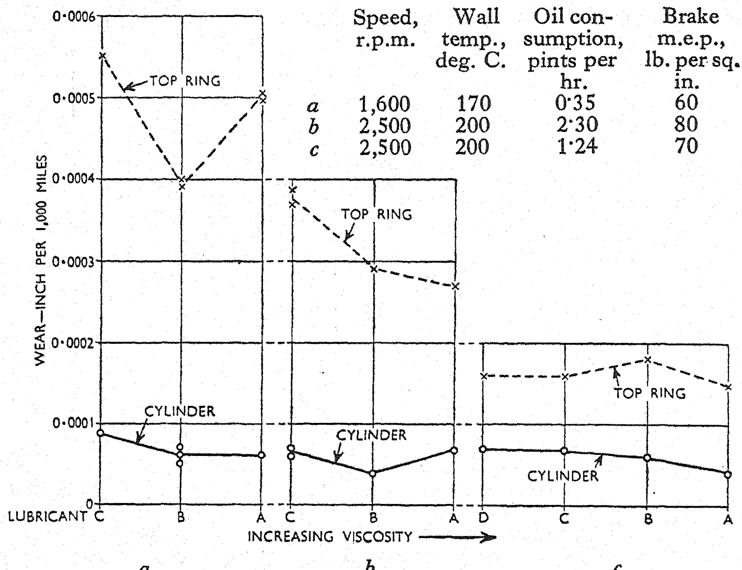
the whole range, while cylinder wear with the heaviest oil was 0·00004 inch per 1,000 miles and with the lightest oil 0·00007 inch per 1,000 miles, but it will be appreciated that these small rates of wear are very difficult to measure accurately.

The results of experiments involving frequent stopping and starting, using lubricants B and D, are given in Table 1 :—

TABLE 1. WEAR WITH FREQUENT STOPPING AND STARTING

Lubricant	Wear, inches per 1,000 miles	
	Cylinder	Top ring
B (medium viscosity) . .	0·0014	0·0143
D (very low viscosity) . .	0·0033	0·0290
B (medium viscosity) . .	0·0015	0·0137

The above figures indicate that, under conditions of operation involving fairly severe corrosion, the very light oil D gave approximately twice the wear of the medium-viscosity oil B.



	Speed, r.p.m.	Wall temp., deg. C.	Oil consumption, pints per hr.	Brake m.e.p., lb. per sq. in.
a	1,600	170	0·35	60
b	2,500	200	2·30	80
c	2,500	200	1·24	70

Fig. 2. Hot-running Experiments

Subsequent experiments, under similar stopping and starting conditions but involving a much lower rate of wear, indicate much less difference between the rates of wear observed with the various lubricants. For example, data obtained with oils A, B, and D are given in Table 2:—

TABLE 2. WEAR WITH FREQUENT STOPPING AND STARTING

Lubricant	Wear, inches per 1,000 miles	
	Cylinder	Top ring
A (high viscosity) . .	0·000044	0·0021
B (medium viscosity) . .	0·000047	0·0033
D (very low viscosity) . .	0·000047	0·0023

The above figures certainly do not suggest that, under these more moderate conditions, the ultra-light oil D showed any greater wear than the heavy oil A.

The foregoing experiments do, however, suggest the possibility that, while the various lubricants are almost of equal merit in so far as abrasion conditions are concerned, the low-viscosity oils may not be so effective in withstanding severe corrosion conditions. Experiments were, therefore, carried out under continuous cold-running conditions and the results of tests on oils B and D are summarized in Table 3:—

TABLE 3. TESTS UNDER CONTINUOUS COLD-RUNNING

Lubricant	Wear, inches per 1,000 miles	
	Cylinder	Top ring
B (medium viscosity) . .	0·00023	0·00066
D (very low viscosity) . .	0·0012	0·0037

The increase in wear with the light oil D is very marked, being in the ratio of about 6/1, therefore tending to support the suggestion that the lighter oils are less able to withstand cold-running or corrosion conditions than are the heavier oils. It could be inferred that viscosity affects the thickness of the oil film giving protection against corrosion. It must be emphasized, however, that the higher wear under corrosion conditions was obtained with an ultra-light oil which is unlikely to find general adoption in this country. In addition, engine operating conditions were such that the oil was supplied to the cylinder bores with practically no delay after starting, whatever oil was used.

It would be anticipated, however, that an important advantage of light oils would be their more prompt arrival at the cylinder wearing surfaces, and experiments were undertaken to obtain quantitative data on the magnitude of this effect. The experiments were carried out on

a $1\frac{1}{2}$-litre six-cylinder engine which was mounted, with the cylinder head removed, in a refrigerator box, the engine being coupled to a 20 h.p. motor.

Preliminary experiments indicated the difficulty of studying visually the spread of the oil film on the cylinder walls, and a special arrangement was, therefore, devised. The principle of the method was to pass an electric current from the cylinder walls to one of the pistons and to measure the voltage drop at the rubbing surfaces. Preliminary experiments had shown that, when motoring an engine, the electrical resistance between a piston and its cylinder bore was considerably greater when an oil film was present than when the rubbing surfaces were dry with

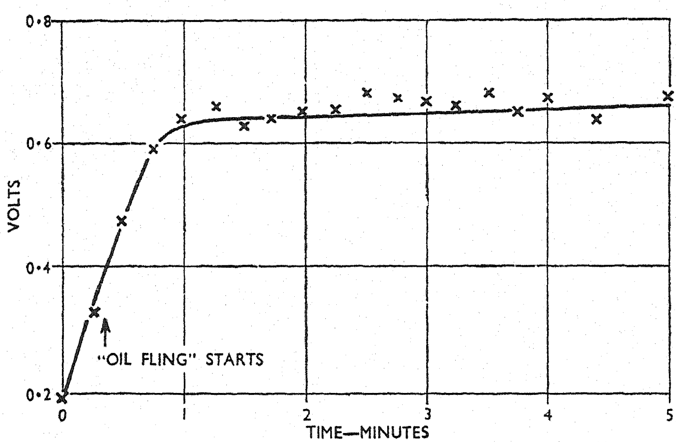


Fig. 3. Speed of Arrival of Light Oils at the Wearing Surfaces of the Cylinder

Oil B; oil sump temperature, 8 deg. C.; motoring speed, 1,000 r.p.m.

relatively good metallic contact. The gudgeon pin of one of the pistons was insulated electrically from its connecting rod, a current of approximately 0·7 amp. was fed from the 240-volt mains to the cylinder block, returning via a flexible lead attached to the piston crown, the lead being supported by a simple linkage arrangement.

At very low motoring speeds the voltmeter indicated cyclic variations in voltage drop, the latter being a minimum at the dead centres. In fact, the voltage drop with the piston stationary was practically the same whether an oil film was present or not, suggesting that, when stationary, the piston rings, etc., penetrated the oil film. The motoring speeds employed in the following experiments were, however, sufficiently high to cause the voltmeter to indicate a fairly steady mean reading. A hole

drilled in the side of the crankcase enabled the approximate time at which oil was first flung from the big end to be noted visually. A typical test record is shown plotted in Fig. 3. It is, perhaps, unlikely that the rise in voltage bears a simple relation to the area of cylinder walls wetted by lubricant, but for comparison it is fairly safe to assume that the attainment of a final steady voltage corresponds with fully wetted cylinder walls, and the time at which this occurs has been quoted in the test results.

Experiments were carried out on the three oils A, B, and D, and at temperatures ranging from -16 deg. C. to $+20$ deg. C. Before each test the piston was withdrawn from the cylinder, and the cylinder walls, piston, and rings were thoroughly cleaned with benzene. The results

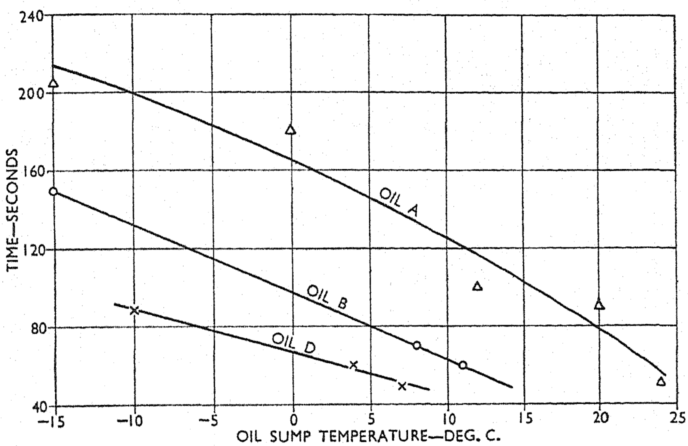


Fig. 4. Speed of Arrival of Light Oils at the Wearing Surfaces Cylinder, in Relation to Temperature

obtained at a motoring speed of 1,000 r.p.m. are shown in Fig. 4, the time for the voltage to reach its final steady value being plotted against the temperature of the oil in the sump. With each of the three oils, the time to establish an oil film increased with reduction in temperature; e.g. with oil B a reduction in temperature from 15 to 0 deg. C. increased the time from 45 seconds to 98 seconds. The results emphasize the beneficial effect of low-viscosity oils. For example, at 0 deg. C. the time to establish an oil film with the three lubricants A, B, and D, was 165, 98, and 67 seconds respectively. It appeared that, with the heavier oils, a large proportion of the time taken to form an oil film at low temperatures was attributable to the delay in "oil fling" from the big end. For example, with the high-viscosity oil A at

-10 deg. C. the total time taken to form an oil film was 200 seconds, of which 120 seconds was the time which elapsed before the first signs of oil fling were apparent. The light oil D showed a lag of only 30 seconds at -10 deg. C. before oil fling commenced.

The foregoing experiments therefore suggest that the somewhat lower corrosion resistance of light oils may be offset by the more prompt arrival of the lubricant on the cylinder walls. The net result on cylinder wear cannot be predicted at this stage, but would be a matter for experiment in each particular case.

The author acknowledges his indebtedness to the Research and Standardization Committee of the Institution of Automobile Engineers, which gave permission to publish these results, and to the staff of the Research Department.

LUBRICATION OF STATIONARY OIL ENGINES *

By Eng. Lieut.-Commander A. Cyril Yeates †

Cylinder Lubrication, Arrangement of Piston Rings, Liner Wear and Lubricating Oil Consumption

The degree of finish on the liner surface, that is, whether ground, or honed, or left with a tool-finish, does not appear to have any appreciable effect on the consumption of lubricating oil. However, it has a considerable influence on the reliability and sensitiveness or otherwise to the grades of lubricating oil that can be used. In spite of the common practice of finishing liners by honing, it is becoming increasingly evident that liners left with a tool-finish are the more reliable. They are less prone to piston seizures and are not in the least sensitive to varying grades of lubricating oil. Experience over a number of years and with a variety of cylinder sizes and operating conditions indicates that a tool-finish to the liner facilitates the lubrication of the piston skirt and reduces the danger of rupture of the oil film.

Pressure-ring form appears to have but little influence on lubricating oil consumption. The clearance of the ring grooves is, however, of far greater importance and the maintenance of this clearance within well-defined limits so as to maintain the oil seal is essential. If this clearance increases to such an extent that the oil seal cannot be maintained, then the rings tend to act as pump valves, delivering oil to the combustion end of the piston, ultimately causing the rings to stick in their grooves.

For low- and medium-speed engines, the most satisfactory method of feeding oil to the cylinders is by means of the controlled mechanical lubricator. With this system, the oil can be introduced to the cylinder at the exact point required for efficient lubrication of the piston rings independently of the skirt. The skirt should receive a copious supply of oil, partly controlled and partly from splash, which it is almost impossible to avoid with the normal trunk piston arrangement. The pressure rings should be protected from excess oil due to splash by means of a wiper ring placed immediately below them, and in this way the lubrication of the pressure rings can be maintained virtually independent of that of the piston skirt. With such an arrangement, the piston rings will operate almost indefinitely without trouble from

* The following notes are intended to apply only to cylinders over 12 inches in diameter.

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sticking, provided that the groove clearance is maintained within the desired limits.

With higher rotational speeds, however, this method of lubrication cannot always be conveniently applied, and the amount of splash from the main lubricating system is more than sufficient to supply all the oil required for piston skirt and ring lubrication. In such cases it is necessary to reinforce the wiper ring defence against excess oil passing the piston. It is advisable to protect the pressure rings by a wiper ring immediately below them as in the previous case, but in order to

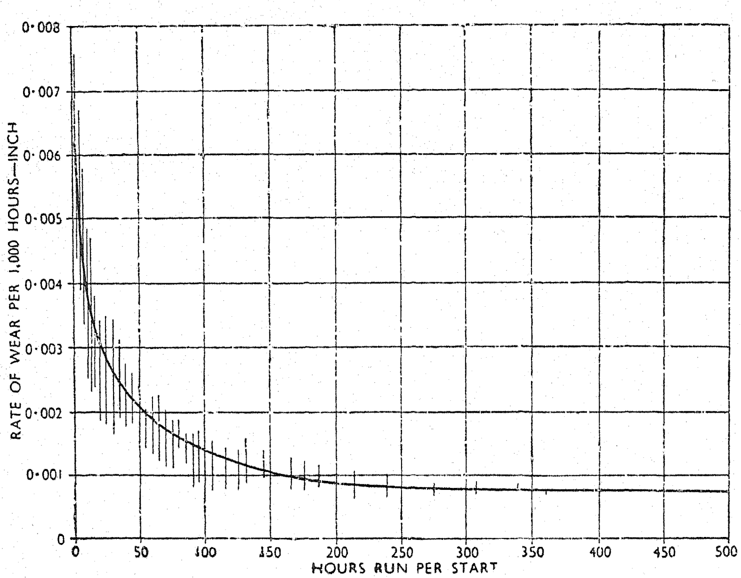


Fig. 1. Effect of Number of Hours' Run per Start on the Rate of Liner Wear

reduce the quantity of oil to be handled by this ring, it is usually necessary to fit a similar ring near the mouth of the skirt.

The rate of liner wear is influenced by a number of factors, such as impurities in the air, lubrication, disposition and fit of piston rings, and effectiveness of cooling, particularly round the combustion end of the liner. There is, however, another factor which under certain conditions appears to have a greater individual effect than any or all of the above. This cause of liner wear may be referred to as "the number of hours run per start," and from the information so far collected, there is evidence that it is influenced to some extent by the characteristics of the lubricating oil used. The effect of the number of hours' run per

start on the rate of liner wear is illustrated in the accompanying curve, which shows:—

(1) That above 100 to 150 hours' run per start the change in the rate of liner wear is neglible.

(2) That below 50 hours' run per start, a very rapid increase in the rate of liner wear takes place.

A possible reason for this effect on the rate of liner wear is condensation during cooling after shutting down, and consequent rusting of the surface; consequently the wear chiefly takes place when starting up, when the rust on the surface is removed by the piston rings; the more frequently this occurs the greater the rate of wear. This may also account for the frequent experience that one, or perhaps two, liners in an engine show a considerably higher rate of wear than the remainder, because the operator always "bars" the engine into the same starting position and thus leaves the same liner exposed on the suction or exhaust stroke. If this premise is correct, the type of lubricant necessary to reduce this effect is one that, under the condition prevailing after shutting down an engine, will adhere to the exposed surfaces and so reduce the tendency for the formation of rust. To a certain extent, the effect can be reduced by barring the engine round one or two revolutions half an hour or so after shutting down so as to ensure that the surfaces are smeared with oil. This same effect also applies, but to a much lesser extent, to the wear of piston rings. These, not being exposed, do not rust, but they have to run over the comparatively rough, rusty surface when starting, and this increases their rate of radial wear.

Main-Bearing and Small-End Lubrication

Gravity Feed. After reliability, the most important feature of any method of applying the lubricant is that of economy. The method of applying the lubricant to the bearings, particularly the small ends, has an immense influence on lubricating oil consumption.

For moderate- and low-speed engines, one of the most successful and economical systems of applying oil to the main bearings is by gravity feed and "banjo" rings to the big-end bearings. Such a method may appear somewhat out of date, but in practice it has been found both reliable and very economical. The usual arrangement is to deliver oil by means of a pump to sight-feed boxes arranged in a convenient position for control. In some designs, worsted wicks, fitted in the outlet from the sight-feed box to the bearing, act as filters and, if the supply from the pump fails for any reason, they continue to feed the bearings with oil for at least an hour or more

after the failure of the feed. With such an arrangement it is possible, in an emergency, to feed the sight-boxes by hand until it is convenient to shut the set down, or until repairs can be effected. This arrangement therefore has the advantage over other systems that failure of the main supply does not necessarily mean an immediate stoppage of the set.

This method of oil feed to bearings has proved satisfactory for journal speeds up to 800 ft. per min. Lubricating oil consumption as low as 10,000 b.h.p.-hr. per gal., based on the rated output of the engine, has been achieved in many instances under normal service conditions, but an average figure for this method is about 8,000 to 8,500 b.h.p.-hr. per gal. for engines running between 200 and 300 r.p.m.

One reason for the economy of the method is the comparatively low pressure of the feed: the oil is not forced out from the ends of the bearings in the form of a fine mist or spray, consequently, the amount of oil splashed or thrown into the cylinders is limited and, because it is not finely broken up, a much smaller area of oil is exposed to the air.

Pressure Feed. Pressure feed to the main bearings, big-end, and small-end bearings is most commonly employed, and with totally enclosed moderate- or medium-speed engines it is essential, as a means both of lubrication and of dissipating the heat from the bearings. As compared with the previous system, it has the following advantages:—

(1) The bearings can be run with finer clearance.
(2) It is less sensitive to temperature changes.
(3) It is capable of utilizing a wider range of grades of oil, particularly as regards viscosity.

It has, however, the disadvantage that, owing to the pressure, the oil is thrown out in the form of a very fine spray, so that considerable quantities are thrown into the cylinders unless special precautions are taken to protect these as far as possible from direct splash.

A feature which has probably a greater influence on the lubricating oil consumption than anything else, is the supply of oil to the small end under pressure. In order to keep the consumption within reasonable limits, it is necessary to reinforce the wiper ring protection on the pistons, and to seal the openings on the side of the piston skirt accommodating the piston pin against leakage of oil from inside.

With large slow- and medium-speed horizontal engines, however, it is not necessary to apply forced feed to the small ends, a fact which has a considerable influence on the oil consumption and the simplicity of the protective arrangements. With this method of lubrication, lubricating oil consumptions as low as 9,000 b.h.p.-hr. per gal. have been obtained in service; a fair average would appear to range between 6,000 and 7,000 b.h.p.-hr. per gal.

WEAR OF CYLINDERS AND PISTON RINGS

By Horace J. Young*

Bearings. Any engine bearing works badly when faulty in (1) design, (2) oil supply, (3) lubricant, or (4) quality of bearing metals. Regarding the last-named, it has been shown (Young 1935) that certain reciprocating bearings, namely, the top end brasses of marine engines, which gave constant trouble over many years, would function successfully on substituting a good bearing metal in place of one which was not good. Further, the main difference between the bad and the good metals was one of *structure*, the successful structure being that which was able, under working conditions, to form and maintain a surface easily wetted by oil and retaining an oil film upon it.

A piston ring and a cylinder wall form a reciprocating bearing which functions satisfactorily from one end almost to the other end, at which extremity it wears. If the rate of wear was no greater than it is over the major portion of the surface covered, this paper would be unnecessary. Cylinder wear, then, occurs at its maximum on that area of the cylinder wall exactly facing the uppermost free piston ring when at its top position nearest to the compression zone. The effect is less opposite the ring immediately below and fades away to little of consequence beneath the lowest compression ring at its topmost position. The circumstances under which, in the author's experience, the rate of wear at these points is more, or less, will be considered.

Saturated Steam Engines. The introduction of superheat to marine engines immediately caused cylinder wear so acute that it became necessary to discover a type of cast iron for cylinder liners and piston rings which would make a commercial proposition of engines converted from saturated to superheated steam. It was possible to do this successfully (Young 1921). Superheated steam engines still suffer from more cylinder wear than is liked, though the trouble with their piston rings was solved. These problems led to the introduction of a series of all-pearlitic cast irons of bearing metal type (Young 1921, 1923).

Dust Contamination. Severe cylinder wear is experienced on engines working in atmospheres carrying impalpable dust, as in districts having dust storms, quarries, cement works, dirt tracks, and so on; and, to a

* The Sheepbridge Stokes Centrifugal Castings Company, Ltd.

lesser yet important degree, in large cities. In J. Sharples's study of the power of various minerals to " take a grip" when drawn by hand over metal surfaces of various hardnesses—the "grip" being preliminary to causing a visible scratch—he found that quartz *just* "grips" a metal with a Brinell hardness number of not more than 650, while topaz behaves likewise to one with a Brinell hardness number of 1,000. Ellis (1937) showed that a soft mineral has wearing effects out of proportion to its position in the hardness scale. Insoluble residues from used engine oils contain oxides, silicates, sulphates, or chlorides of iron, aluminium, calcium, magnesium, silicon, and sodium, each (with carbon) altering in amount, on any one engine, according to environment and fuel used. The predominance of a particular dust necessitates a particular liner metal, which may not be the same as that needed economically to resist another composition of dust.

Cylinder Design. A greater thickness of that part of the cylinder wall facing the top position of the upper rings, when causing increased heat sufficient to harm the oil film, leads to a higher rate of wear. In tractors burning paraffin, one design incorporated extreme "siamesing", resulting in the whole of the wear, which was excessive, being confined to the hotter side of each cylinder. Engines of some vessels passing through the Persian Gulf, where the cooling system is probably 30–50 deg. F. above normal, have suffered more rapid wear than when operating in temperate climates.

Any factor causing excessive heat at the top of the cylinder increases wear by destroying the oil film; and undue wear, apart from that caused by abrasive dust, occurs when contact is established between the material of the ring and that of the cylinder wall. The rate of wear is low because moments of contact are rare, otherwise, excessive wear would happen in a few hours.

Stopping, Starting, Manœuvring, Idling. The total number of times the piston reciprocates over a given number of hours or miles of running provides no indication of the total cylinder wear experienced. The cylinder is not worn at a rate governed by the *amount* of use. Services involving many stops of long duration suffer a rate of wear higher than those engines having stops which are no fewer but are of short duration, insufficient to allow the oil film to drain or the system to cool or, if the engine idles, to permit of local overheating.

Excessive heat and excessive coolness increase cylinder wear for the same reason, namely, lack of oil film at the position where wear occurs. The oil cannot maintain itself as a perfect film at the top position of the topmost ring when (*a*) the cylinder walls are cold and

(*b*) the oil itself is cold. Further, if the top of the cylinder was very hot when the engine was stopped the condition of the oil film thereon will be parlous after being allowed to drain and cool.

When an engine frequently labours there is increased bore wear. A high-powered car suffers less wear than a low-powered car with the same body weight; and a single-decker bus than a double-decker, even though on the same service, particularly when a hilly one. "Flutter", blow-by, overheating, deposits inside or outside the liner, overcooling, thrust, petrol wash, dilution, poor combustion and any conditions putting over-stress upon, burning, cooling, draining, or weakening the oil film, or reducing the supply of oil to the "high water mark", lead inevitably to moments of contact between the top ring and the wall.

Speed of Piston Travel. High speed affects the whole area and both extremes of the area travelled and, by itself, does not cause increased cylinder wear. Very low speeds may not be so innocuous.

Frictional Grip. On railways it is found that the lower the peripheral speed of a wheel the more intense the frictional grip between rim and brake. Viewing the forces operating upon the top piston ring as the piston approaches and passes the upper dead point, the above phenomenon may come into play when the oil film is imperfect and the piston speed low.

Surface Finishes. When the Brinell hardness number of the metal of the cylinder or liner is below about 850, it forms in time its own working skin; hence the running-in of such irons is a period when their surface is forming. Bearing metals come out of service with a skin totally different from their machine-shop finish; they possess the power to form (and to re-form, when damaged) a surface such as is wetted by oil as an unbroken film. The ideal finish would consist of an amorphous skin. In large engines, particularly those using superheated steam, extremely rapid bore wear, ring wear, or breakage occurs if, on reboring the cylinder, ferritic iron is exposed (Young 1935).

Corrosion by Oil. The author has shown (1927) that used oil from motor-ships and automobiles sometimes is corrosive and that chemical tests fail to detect it. This finding was unpopular, though Taub has stressed its importance recently (1937). Diesel exhaust gases (from fuel containing 1 per cent of sulphur) contain about 0·04–0·05 lb. of sulphuric acid per 1,000 cu. ft.; moreover, if the air (from a "wet"

compressor using compound lubricant) passes through copper intercooler tubes it gathers corrosive copper salts.

The degree of attack on liners by contaminated oil is infinitesimal, but it lowers the fatigue resistance of the skin of the iron. Wear starts with the weakening and loosening of one particle, causing over-stress of adjacent particles, fatigue sets in, the particles loosen, disappear, and the iron has "worn". Machine-shop finishes stress the surface layers, affecting the "life" of the liner to a degree yet unexplored. Prof. Sawin (1937) of Skoda Works states that wet and dry grinding decrease resistance to wear at a depth up to 0·05 mm., and that wear resistance bears no known relationship to chemical analysis or to hardness of materials not of the same origin and structure, all of which the author's experience confirms.

Wear by Rusting. The Institution of Automobile Engineers found that wear increased with wall temperatures kept below 80 deg. C., and this emphasizes the need for a lubricant which behaves normally under supercooled, as well as under normal, conditions. The following phenomena do not agree with a theory of severe rust attack:—

(1) Major cylinder wear occurs on the hottest part of the cylinder.
(2) Some cylinders of engines on "cold" services (taxicabs, etc.) are worn on one side only.
(3) Cylinders worn all round do not show double the diametrical wear of those worn on one side only, but give similar results.
(4) Saturated steam cylinders are always wet, sometimes with brine; their wear is low.
(5) The wear from cylinder to cylinder of one engine varies; often the worst cylinder is not the same one in every engine.
(6) Marine Diesel and automobile cylinders opened for inspection are unremarkable for the presence of rust; non-rustable liners are unremarkable for its absence.

H. E. Smith (1937) has found that high atmospheric humidity is associated with a low rate of wear. Southcombe (1928) states that water slowly displaces oil from metal surfaces even under static conditions. The author has seen ships' engines unaffected by more water in the lubricant than anyone unfamiliar with marine work would credit and thinks that the displacement under static conditions only is important to this present subject.

Analysis and Structure. Bearing qualities depend more upon structure than composition. The analysis of an iron is a measurement of its contents, while the microstructure is an "aerial" view showing how those contents are arranged.

Brinell Hardness and Tensile Strength. The best-wearing grey iron liners which have come before the author have been neither hard nor very high in tensile strength. When sheer hardness is relied upon it should exceed a Brinell hardness number of 850 to be efficacious.

Low Carbon; High Phosphorus. Years ago a high-class low-carbon iron obtained wide popularity. It wore unduly (Young 1926, 1928). The author's experience is that the carbon content should be as high as the texture of the particular iron can hold safely; hence the importance of structure. Irons containing ferrite are improved by more phosphorus (Young 1937), but the permissible amount is governed by the thickness and mass of the casting concerned (Young 1934).

Grain Size. "Close-grained" cast iron is a fetish. Very good liner irons, including "Lanz Perlit", have larger grain than ordinary irons. Little is known about the effect of grain size; it has been a superstition like that concerning sulphur (Young 1921).

Alloys; Superimposed Coatings. The learned use of alloys is progressive, their incorporation in cast irons of the bearing-metal type being good when imparting further resistance to stress, heat, corrosion, or oxidation *without harming the microstructure.*

Superimposed coatings sometimes break down through fatigue causing separation, flaking, etc.; the period before this occurs being variable and inestimable. The problem is to find commercial methods of testing the thickness, perfection, and adhesion of the coating *before* passing *each* article for service.

Piston Rings. The topmost compression ring spreads the oil as a film over that portion of the cylinder wall over which it alone travels, the surface layers of which are considerably hotter than those of the ring. The ring is confined in the ring groove, excess heat there being transferred to the back of the ring with consequent carbonization of the oil film, sticking of the ring, etc. That it is "better to wear the ring than the cylinder and, therefore, to use soft rings", is a costly fallacy. The hardness or toughness of the ring should more than equal that of the wall; actually, may be far greater without ill effect. When ring wear is troublesome, on an otherwise normal superheated steam or internal-combustion engine, the above remedy reduces it and, certainly, does not increase bore wear.

Heat. Fluctuations of heat entering a liner affect, as the sun's warmth

affects an ocean, the surface layers only. Those of the order of 11 deg. C. at 0·5 mm. depth become of the order of 3 deg. C. at 1·5 mm. and disappear altogether at 5 mm. The temperature and the fluctuations in the skin of the iron, say, at 0·001 mm. depth are unknown quantities; the author believes they are great and produce stresses leading to fatigue breakdown of the skin, particularly just after an engine is started from cold and while no heat gradient to the water exists.

Sulzer (1926) showed that the piston rings of a two-cycle marine Diesel engine were cooler than those of a steam-jacketed steam engine, the top ring of the former attaining a maximum of only 126 deg. C. (259 deg. F.) when the liner had a maximum temperature of 253 deg. C. (487 deg. F.) and the piston crown (midway between centre and circumference) had 211 deg. C. (412 deg. F.), all measurements being at 0·5 mm. depth. Overheating vitally concerns two areas, namely, the surfaces of the upper part of the liner and of the top piston ring grooves, one effect being always destruction of the oil film followed by wear of liners or grooves, stuck rings, etc.

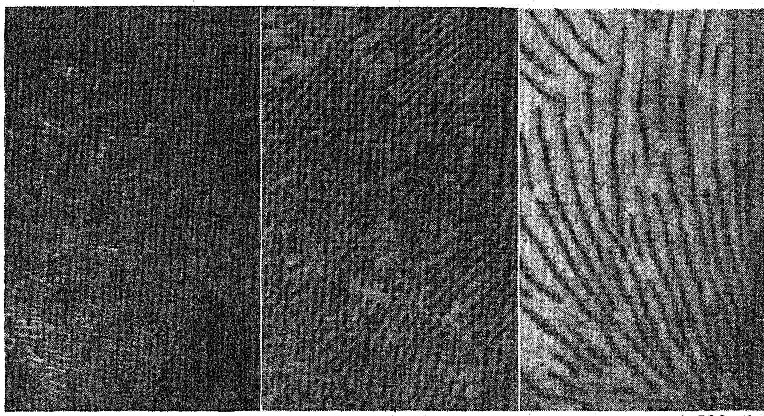
Metallurgical Aspect. Dry sleeves of superior cylinder iron improved by centrifugal casting are used largely for automobile engines; they are likely gradually to replace reboring. For automobile and other engines on commercial or important services the selection of the correct liner material is a question of economics. In some cases the liner need not resist satisfactorily for longer time than the predetermined date when the engine comes in for overhaul. In others the liner has to run the maximum time without renewal. There are qualities to meet all requirements. The abrasive-resisting part of the demand is met specially by the nitrogen-hardened iron liner of over 900 Brinell hardness number, with which the work of J. E. Hurst is so closely associated. Austenitic irons and chromium-plated products are available for corrosion-resistance; all-pearlitic irons for bearing-metal properties.

The last-named have always been the author's study and practice for rings and liners of all engines, from Diesels of 1,000 h.p. per cylinder to motor cycles. The first large liners for superheated steam were made of an early type of this iron (Young 1921). In 1924 came the greatest impetus to our knowledge of pearlitic iron in the Lanz Perlit process (Young 1925, 1926); an all-pearlitic iron developed by the author was introduced in 1933 (Young 1934, 1935) and a new series is being perfected.

A bird's-eye view of all-pearlitic iron may be obtained from the following scale:—

Structure of the iron	Engineering quality
(1) Pearlite with considerable ferrite. .	Never does well.
(2) Pearlite with traces of ferrite . .	Never does *very* well.
(3) Solely pearlite (no ferrite or cementite).	Theoretical all-pearlitic iron.
(4) Pearlite with traces of cementite . .	All-pearlitic iron as manufactured.
(5) Pearlite with considerable cementite .	Inclined to be hard.

The fourth on the scale is the correct product because its content of a trace of cementite is a guide enabling the producer to keep the iron just on the safe side of the theoretical all-pearlitic point. This fourth quality has countless possible variations of composition, also of the pearlitic structure itself. As examples of the latter, Fig. 1 depicts at



×1,500 dia.

Fig. 1. Three Types of Pearlite

the same magnification and in castings of similar mass and thickness, three of the many kinds of pearlite. The type of pearlite present in an iron has a profound effect upon its qualities.

Conclusions. There are no mysteries about bore wear, only difficulties. Every bearing requires oil and a bearing metal; and none, especially a reciprocating one, can be expected to wear normally when one portion of its travel sometimes lacks oil film or a film free from powders and is exposed always to fluctuations of heat together with superheated steam or the gases of combustion. Much can be done to get oil to the top position but this, to be successful, also needs improved metals for the barrels, namely improved bearing metals possessing, also, the power to resist abrasive action and fatigue, whether caused by dust, by the destruction or absence of the oil film, or the action of corrosive oil and gases.

Skin fatigue, the author believes, is the cause of the start of particles disappearing, i.e. of wear; this fatigue is not only induced by scoring during dry or semi-dry moments of contact, but by etching attack, heating and cooling stresses, and stresses left by machine-shop finishes.

In this paper it has been possible merely to indicate the outstanding points of data collected over several years. The author has tried to avoid opinions, to present facts backed by observance of happenings in engines in innumerable spheres of action and to keep within the boundaries of his own profession, metallurgy. He desires to thank Mr. Frank W. Stokes and the directors of the Sheepbridge Stokes Centrifugal Castings Company, Ltd., for permission to give this paper.

References

ELLIS, O. W. 1937 Jl. Inst. British Foundrymen.

SAWIN, N. 1937 *Machinery*, May.

SMITH, H. E. 1937 Congress Book, London Congress, International Association for Testing Materials.

SOUTHCOMBE, J. E. 1927–8 Trans. North East Coast Inst. Eng. and Shipbuilders, vol. 44, p. 207.

SULZER, R. 1926 Trans. Inst. N.A., vol. 68, p. 26.

TAUB, A. 1937 Engineering Soc. City and Guilds College.

YOUNG, H. J. 1921–2 Trans. Inst. Mar. Eng., vol. 33, p. 659.

1923 Trans. Inst. Mar. Eng.

1925 Jl. Inst. British Foundrymen.

1925 Manchester Metallurgical Soc.

1926 Jl. Inst. British Foundrymen.

1926 Jl. West of Scotland Iron and Steel Inst.

1926 Diesel Engine Users' Assoc.

1927 Trans. Inst. Petroleum Technologists.

1928 Jl. Inst. British Foundrymen.

1934 Trans. Inst. Mar. Eng.

1935 Proc. Inst. Automobile Eng.

REPORT ON PAPERS IN GROUP II

ENGINE LUBRICATION (INTERNAL COMBUSTION ENGINES)

By H. R. Ricardo, B.A., F.R.S., M.I.Mech.E. (*Member of Council*)

There are more than thirty papers in this group dealing with the lubrication of internal combustion engines, and several in the other groups which have some bearing on the matter. In the short space available for this summary, it is not possible to give an adequate review of each individual paper. It is thought best, therefore, to consider in turn the major problems which come under consideration, indicating, in each case, the points upon which there is general agreement and those on which there still remains uncertainties. The latter, it is hoped, the discussion will help to clear up.

The most popular and therefore, perhaps, the most painful subject, is that of cylinder bore wear. This brings us into the thick of the battle between the corrosion and abrasion theories. That both effects occur, there is little doubt, but there is considerable difference of opinion as to their relative importance. Much of this divergence appears to be due to differences in the operating conditions, and in this connexion, Fig. 1 of Yeates's paper is very interesting. This diagram, which gives wear rates apparently collected from large oil engines in service, shows a close relationship between the wear and the running period between starts. If the horizontal scale of this diagram is inverted, so as to read "starts per 1,000 hours" instead of "hours per start", the original curve becomes roughly a straight line; thus the wear may be expressed as so much per hour run plus so much per start, and the slope of the line shows that each start causes as much wear as about sixteen hours' running. Thus, in the typical case of eight hours' run per day, two-thirds of the wear appears to be due to starting and only one-third to running.

Some figures recently obtained on sleeve valve engines tend to confirm this conclusion. Thus, of two similar engines, one running 8 hours per start, the other 170 hours, the former had a wear rate about three times that of the latter, a ratio which agrees fairly well with Yeates's curve.

On the other hand, the tests of Everett and Keller on automobile engines show only a small starting wear (apart from that due to bedding-down of disturbed rings), equal perhaps to one hour's running. Part of this discrepancy may be due to the quicker warming-up of the small

engine, and some, perhaps, to differences in materials, but it seems unlikely that such a large difference can be completely explained in this way. It may be that the corrosive products formed when an oil engine is started are more harmful than those of the petrol engine.

Taub and Young do not consider that corrosion is normally the main cause of wear, the former considering it to be the result of blow and best remedied by the fitting of improved piston rings and by a plentiful oil supply. Young cites a number of facts telling against the corrosion theory, and makes the interesting point that, in saturated steam engines, where the bore is always wet, little wear takes place, although the conditions favour corrosion. It may be doubted, however, whether the condensed water contains enough gas to be corrosive. Williams, in an earlier paper, has shown that these gases are important factors.

As regards cylinder materials, there is a fair measure of agreement. Ottaway, Pearce, and Young all agree on the advantages of pearlitic structure and of high phosphorus content. Several authors also mention the advantages of austenitic iron, at any rate under cold conditions, and of a coarse graphitic structure.

Coming to lubricants, Williams shows that there is an appreciable viscosity effect under cold conditions, low viscosity giving high wear. This, however, is to some extent offset by the more rapid arrival at the bore of thin oil after starting. Under hot running conditions, viscosity has little effect. As regards composition, Bass, Bouman, and Norlin agree that "oiliness" dopes have little effect, but Rosen shows that anti-oxidation dopes reduce wear considerably.

The value of graphite as a running-in compound is generally admitted, but opinions differ as to its value for normal running. Higinbotham quotes evidence in its favour, but Norlin's road tests suggest an increase in wear due to its use. Road test results, however, are inevitably erratic, as Norlin's figures reveal, and in this case the number of tests seems insufficient to give a fair average.

Turning to other groups, Finch and Zahoorbux give a very interesting account of the running-in process and its effect on the atomic structure of the rubbing surfaces. It appears that running-in profoundly changes the surface condition, and further study of this effect may provide a new line of attack on the wear problem.

Another thorny subject is that of carbonization, ring sticking, and other such evils arising from decomposition of the oil. Luckily, the worst trouble, ring sticking, has recently been met by the introduction of the tapered ring. In the reporter's experience, this has proved to be an almost certain cure for engines up to about 7 inches bore. The reporter has, however, no evidence regarding large engines; if there is any he hopes the discussion will produce it.

The treatment which the oil undergoes in the engine is so complex and variable that one cannot be surprised at the conclusion reached by Barton, Bass, Bouman, and Kadmer, also by several authors of Group IV papers, that laboratory tests are not in general safe guides to the behaviour of the oil in the engine. In a Group IV paper, Moutte, Dixmier, and Lion point out that the oil alternately suffers low-temperature "stewing" in the crankcase, and high-temperature "frying" in the cylinder, and describe a method of test which incorporates both these conditions. This test, they claim, gives results which match more closely those of engine tests than do simpler methods. Rosen also describes a new form of laboratory test which he has found to give fair correlation with engine results.

One point on which laboratory and engine tests do seem to agree is the improvement due to anti-oxidation dopes, as will be seen from comparison of Hanson and Egerton's paper with Rosen's statement quoted above, to which the reporter himself can add some confirmation.

Some interesting points that deserve mention are:—

(1) Freeman has found that large additions of new oil to old increase sludging, so that make-up oil should be added gradually. It would be interesting to know if others have had the same experience.

(2) Flowers suggests that the fine carbon in used oil has the beneficial effects of colloidal graphite. It may be that this carbon, or some of it, is actually in the graphitic state, a question which could presumably be settled by X-ray analysis.

(3) Jones and Turner quote figures which show that the large improvement in oxidation resistance, obtained by modern refining methods, is not accompanied by any loss in lubrication quality.

(4) Auld and Nicholson point out the importance of minimizing aeration in the circulating system, particularly in engines with oil-cooled pistons.

Oil Consumption. On this point there are two schools of thought: the American, represented by Taub, who advocates a plentiful feed to the cylinder with special high-pressure rings to give drastic scraping; and the European, as represented by Ottaway, who states that high-pressure rings give no improvement in prolonged running, and that it is still necessary, therefore, to limit the quantity thrown off from the big end. This, however, is not easy in high-duty engines, where a large quantity of oil must pass through the big end in order to cool it.

Dicksee, incidentally, points out the importance of the location of the big-end oil hole in controlling this flow. Ottaway also advocates a wide top ring, which gives greater initial oil consumption, but wears the cylinder less and thus gives a lower average consumption. Perhaps this could be combined with the tapered ring face, advocated by Taub, which hastens the bedding-in process. Another interesting point mentioned by Ottaway is the need for a deep top land and for piston rigidity. In the latter case, he suggests that flexure of the piston skirt may affect the transference of oil past the piston.

Taylor considers that piston ring lubrication is normally of the boundary, rather than the film, type, and quotes, in support of this contention, figures obtained by motoring an engine with varying gas loads on the piston. He suggests that the sudden increase in oil consumption observed at high speeds is due to a change from boundary to film conditions, whereas Taub ascribes this effect to ring "flutter". It may have some connexion with gas blow, for this often occurs quite suddenly above a certain speed, and the reporter has met a case in which an increase in the oil supply to the bore caused severe gas blow. Whatever the cause, this question of piston ring action at high speeds deserves further study. Possibly the electrical method used by Williams would throw some light on the matter.

Bearings. So far, only the cylinder has been dealt with, and it is time to consider the rest of the engine. Here, the most acute problem is that of materials for the big-end and main-journal shell, especially in high-speed oil engines. Whitemetal is apt to crack under heavy load and lead bronze has several drawbacks, for it is expensive, it needs a hard shaft, and the clearances must be greater, which increases the difficulty of oil control. Moreover, as Bass points out, it is corroded by some oils, the use of which is desirable on other grounds.

Bearing materials have had little attention in this group of papers, but in Group I there are two papers, one by Macnaughtan dealing with tin-base metals, and the other by Neave and Sallitt on copper alloys, which form a good review of the situation.

For big ends, the crux of the problem is cooling. Whitemetals lose fatigue strength rapidly with increase of temperature, but if they can be kept cool enough, will generally withstand service conditions. This argument is supplied by evidence from Williams's paper in Group I. The difficulty lies in keeping the whitemetals cool, since the oil flow is limited by the amount the scraper rings can deal with. One solution is suggested by Taub, i.e. to use a large big-end oil flow and drastic scraping. Another is to circulate large quantities of cooling oil through the interior of the crankpin, allowing only a small feed to the bearing

itself. This has proved successful in some aero-engines with extremely severe big-end loading.

There have recently appeared some improved alloys which are claimed to have better mechanical properties than normal whitemetal at high temperatures, and yet to work well on a soft shaft. Apparently the behaviour of those metals is not discussed in the papers, and the reporter's experience, though favourable so far as it goes, is limited. It is hoped that something further will come to light in the discussion.

Synthetic resin has also been suggested as a bearing material since it has done well in other applications, but the reporter does not know of any case where it has been used in internal combustion engines (except experimentally). It appears to be very "kindly" to the shaft on which it runs, but there is some doubt whether it can withstand the high temperatures of internal combustion engine bearings. Several papers in Group I deal with the use of this material.

Taylor calls attention to another factor which may contribute to the troubles of heavily loaded bearings, namely, flexure of the parts. Considering a big end at firing top centre, it is clear that the loaded face of the crankpin will tend to become concave under load, and the big end convex. If the two have the same curvature, the load will be evenly distributed, otherwise it will be concentrated either in the centre or at the ends. This may account for the erratic nature of whitemetal big-end troubles, some designs being free from them, while others, with apparently similar bearing conditions, fail. It may be that the successful designs are those in which, by good design or good luck, the two curvatures are equal.

Another question raised by Taylor is the effect of load variation on carrying capacity. Little information, if any, is available on this point. Load reversal is certainly an advantage with low rubbing speeds, as is shown by the well-known fact that the gudgeon pin of a four-stroke engine will withstand far heavier loadings than that of a single-acting two-stroke engine, where there is no reversal. Whether it has any appreciable effect at the higher rubbing speeds of big ends and main journals is another matter, and research is certainly needed on this point.

Mickelsen makes the bold suggestion that it may be possible to lubricate a crosshead-type engine with water or a water-oil mixture, as in the old Willans steam engine. The reporter is inclined to believe that cylinders also could perhaps be lubricated in the same way, though not, of course, with pure water. Such a scheme has many attractions. Water is a much better coolant than oil and, moreover, is itself easily cooled, which is more than one can say for oil. Again, the rate of consumption would be of little account, except in aero-engines, if most of the lubricant came from the tap. The presence of water might induce

corrosion, but this difficulty has been overcome, with the water-cooled piston rods of double-acting engines, by the use of inhibitors.

That this is not entirely an idle dream is shown by experience with a car which suffered badly from water condensation in the crankcase. Growing tired of removing pints of water from the sump every few hundred miles, the owner changed to a compounded oil, which formed a stable water emulsion, and simply let the water accumulate. It eventually stabilized at about a "50/50" emulsion with the oil, and the car ran for many thousands of miles on this brew with no apparent ill-effect.

An important feature of emulsions, as compared with homogeneous lubricants, is their viscosity characteristic. The ideal lubricant would have a viscosity independent of temperature. This condition is not approached even remotely by any known oil, but the same does not apply to emulsions, some of which, shaving cream, for example, actually increase in viscosity as their temperature is raised. It will probably, however, be necessary to keep the bearing temperatures from rising to much over 100 deg. C., for the data of Jakeman and Fogg for seizure at temperatures above this figure with oils containing small quantities of water suggest that "steam locks" may impede lubricant flow into the loaded area.

Returning to earth, Barrington and Lutwyche's paper gives some much-needed data on a rather neglected subject, the friction of cold engines under starting conditions. It is rather horrifying to learn that a two-litre engine may need as much as 15 h.p. to crank it at 500 r.p.m. It appears that at the breakaway the piston rings are responsible for most of the friction, but that, once moving, the piston skirts are the chief offenders.

Filtration. Filtration is, undoubtedly, beneficial, for bearing wear at any rate is mainly due to hard particles in the oil, though their share in cylinder wear is perhaps not so prominent.

It is an open question as to whether oil, if properly filtered, can be re-used indefinitely. Beale quotes cases of engines in which the oil has not been changed for ten years, though it is probable that oil consumption and replenishment had, by that time, eliminated all but a minute fraction of the original oil. Young considers that used oil becomes corrosive, but Flowers believes that the increased acidity of used oil is, if anything, beneficial on account of the oiliness it confers. In any case, this acid can be removed, as Cahill and Dolton and Mackegg point out, by washing with water in a centrifugal separator.

Pickard gives some interesting figures as to the size of particles passed by various types of filters. Comparing these with the figures for working

clearance of bearings given by Mickelsen, it appears that, even with filtration, something is left to scratch the bearing surfaces, though the worst offenders are doubtless removed. De Langen shows that magnetic filters can remove extremely small particles of iron and can reduce wear. In cases for which the reporter has figures, these ferrous particles average about one-half of the total incombustible matter in used oil, and it would be interesting to know what part they play in the total abrasive effect.

To conclude, the reporter feels that space limitation has compelled him to leave unsaid many things that he ought to have said, and to omit reference to many useful and interesting points. This particularly applies to those papers which deal in a general way with lubrication practice in one field or another. Such papers, generally from authors having an unusually wide range of experience, are of great value, but are, from their nature, almost impossible to summarize. The reporter hopes that he has not also committed the converse crime of saying those things that he ought not to have said, in the sense of misinterpreting any author's views. If so, he apologizes in advance, and pleads, in extenuation, that in some cases he had not had available the illustrations of the papers when writing this summary. On this account, he had sometimes been uncertain as to the author's exact meaning.

GROUP II. ENGINE LUBRICATION (RECIPROCATING STEAM ENGINES)

By W. A. Stanier, M.I.Mech.E. (*Vice-President*)

Many of the problems of the internal combustion engine are very similar to those of the old reciprocating steam engine, so this report will cover papers which seem to refer to matters affecting both types of prime mover.

It is to be noticed that the French and German State Railways consider that various grades of superheater cylinder oil are desirable according to the degree of superheat obtaining in the cylinder, whereas the Canadian National and English railways employ only one grade. Of the opinions expressed about superheater cylinder oils, the majority favour compounded oils, since it is considered that at the temperature of superheated steam the oil becomes much less viscous and the fatty oil is partly decomposed, the decomposition products helping in the formation of stable and resistant boundary films. Oils A and B, as quoted by the French State Railways, and the valve oil as used by the Canadian National Railways, are generally similar to those in use on the English railways:—

	French State Railways Oil A	French State Railways Oil B	Canadian National Railways	A typical English railway
Specific gravity	0·904	0·903	0·903	0·915
Viscosity, sec. Redwood	186 at 212 deg. F.	165 at 212 deg. F.	168 at 210 deg. F.	165 at 212 deg. F.

For oil C, as used on the French State Railways, with a specific gravity of 0·905 and a much higher viscosity of 257 sec. Redwood, there is no comparable oil on the English railways, but it appears to be somewhat similar to the special high-quality cylinder oils used by the German State Railways for locomotives with high superheat up to 752 deg. F. The three grades of cylinder oil used on the last-named railways appear, from the analytical data, to be straight mineral oils, as follows:—

Grade 1. Specific gravity, 0·903; viscosity, 228 sec. Redwood at 212 deg. F. and an open flash point of 645 deg. F., for locomotives with high superheat up to 752 deg. F. and average piston speeds of 1,370–1,570 ft. per min.

Grade 2. An oil with a lower flash point for locomotives with a high superheat, but lower piston speeds.

Grade 3. Specific gravity, not over 0·95; open flash point not under 572 deg. F.; and a viscosity of 156–219 sec. Redwood at 212 deg. F.

Grade 3 approximates somewhat to English practice.

Of special interest is the use of emulsified oil, prepared by the German State Railways from superheated steam cylinder oil and lime water, for use in locomotives working under medium loads.

With regard to the oils used for journals, motion, etc., the German State Railways use winter and summer grades, as do certain of the English railways, while the Canadian National and the other English railways prefer the same grade throughout the year; one of the English railways considers that the inconvenience of changing the grade of oil twice a year outweighs any possible advantage and in its experience no advantage was obtained when the thicker summer oil was used. It is the practice of the English railways to use a mineral oil containing a certain percentage of refined raw rape oil, the percentage depending on the different classes of work and the experience of the companies concerned, whereas the German State and Canadian National Railways use a mineral oil only. It is interesting to note that the German State Railways use an oil of higher viscosity for lubricating the journals and gear of streamline locomotives, this also being the practice of certain English railways.

It is the practice of the German State Railways to use wick trimmings for supplying the oil to the valve gear, and to the connecting and coupling rod bushes, while the English railways use worsted trimmings for the valve gear and either worsted trimmings, needle trimmings, or felt pads for the rods. When needle roller bearings are fitted to the valve gear in this country, grease is used as the lubricant.

On the modern locomotive, the pump lubricator has generally superseded the hydrostatic system for supplying the lubricant to the cylinders and the general principles employed by the German and French State Railways and the English railways would appear to be similar as regards the position of the pump, the provision and position of back pressure valves, etc. The English railways, however, do not favour the branching of lubricator pipes as indicated by Chatel for use under certain circumstances. A point of interest is the recommendation by Chatel that the high-pressure cylinders should be lubricated towards the dead centres, where the piston is almost stationary, so that the piston rings can receive plenty of oil. On the English railways a feed taken direct to the cylinder barrel is situated in the middle of the stroke.

Most of the French railways inject steam or water into the cylinder

exhaust to reduce the temperature of the gas when coasting, and they have under consideration an automatic device to come into operation when the driver closes the regulator, whereas the English railways attain the same end by allowing a "breath" of steam to enter the cylinder by means of the regulator or through the anti-carbonizer or atomizer.

The analysis and mechanical tests of cast irons for piston rings and cylinders given by Chatel correspond to those generally in practice on the English railways, except for the higher percentage of silicon and the lower percentage of phosphorus, which should be noted. One railway in this country is, however, experimenting with a lower percentage of phosphorus, also with the use of nickel and chromium.

The tendency towards the use of a harder metal for the rings than for the cylinders is interesting: this practice is also advocated by Young in his paper on "Wear of Cylinder and Piston Rings".

Modern French locomotives are fitted with four rings per piston head, the German with five, while the English railways favour two or three, but experiments are now being carried out in this country with an increased number of rings. Piston ring wear varies considerably according to the class of locomotive and the type of work on which it is engaged. For express passenger compound engines Chatel gives mileages of 12,500–25,000 for the high-pressure cylinders and 28,000–37,500 miles for the low-pressure cylinders, at which the rings are replaced. On one English railway the piston rings of a three-cylinder (simple-expansion) locomotive are changed at 40,000–45,000 miles.

It is possible that the general increase in average speeds and distances between starting and stopping points, and the increase in steam pressures and temperatures, have resulted in a running temperature in the cylinder that tends to break down the surface of the cast iron. The general appearance of pistons and rings is such that they seem to have reached a glazed condition, but that this condition has been broken down, apparently intermittently; this may be the cause of the short mileages now reached in the life of piston rings. Experiments are being made, particularly in America, with the use of bronze rings, and also with sectional rings made of bronze and cast iron. It is suggested that this is hardly a lubrication question but more a metallurgical matter, and Pearce indicates how far metallurgy has progressed towards the production of suitable cast iron for cylinders, pistons, and piston rings.

The comparison of oil consumption is difficult, since the consumption is governed by working conditions and is not on an entirely quantitative basis: Nordmann and Robrade give a figure of 0·7 pint per 100 miles for a German "series 03" (two-cylinder) locomotive, the figure for a two-cylinder locomotive on the English railways being 1·0–1·2 pints per 100 miles.

No mention is made by Nordmann and Robrade of the atomization of the oil for cylinder lubrication, a point on which the English railways lay stress, but Chatel mentions that some use is made of atomizers combined with the back-pressure valves.

The tests indicated by Chatel for mechanical lubricators would appear to be applied to a new type of lubricator and not to be service tests, which in the case of English railways require the lubricator to maintain a pressure of 250 lb. per sq. in. for cylinder lubrication and 200 lb. per sq. in. for axlebox lubrication.

In dealing with marine engine lubrication practice, Freeman expresses general satisfaction as to the way in which the oils have been improved to withstand the much heavier duty imposed by modern conditions, but considers the wide variations in price are not commensurate with their performance. Superheated steam valves, pistons, and liners are lubricated by mechanical lubricators feeding into the steam pipe or round the high-pressure liners, the oil used being similar to that generally used in locomotive practice. The turbine oil, mentioned by Freeman, is of the "extra heavy" grade, and is required to demulsify by the method of the Institution of Petroleum Technologists in three minutes; this is a very low figure for such a heavy oil, and it would be interesting to know its specific gravity and ageing properties.

In their paper on oil circulation systems, Auld and Nicholson draw attention to the importance of using oils which are least affected by the joint action of heat and oxidation. Increase in viscosity, development of acidity, sludging, and carbonization are some of the chemical changes in the oil responsible for the lubricant "breaking down" in service. The remedy against these evils lies in the suitable choice of stocks and the subsequent refinement, care being taken that the latter operation is not overdone so as to produce a tendency to acid formation.

Oil circulation systems fitted to enclosed steam engines give rise to difficulties peculiar to themselves. Owing to the liability of a small amount of cylinder oil to find its way into the crank chamber, Auld and Nicholson recommend, on certain types of engines, the collection and examination of oil from the separator to ascertain its suitability for re-use. The height of the oil level in the crank chamber and the pressure of the lubrication system (a maximum of 10 lb. per sq. in. being recommended as sufficient for most engines) are also points to which the authors suggest attention should be paid.

Since aeration of an oil is one of the chief factors in the acceleration of oxidation, it is important that it should be eliminated as much as possible. Auld and Nicholson consider that aeration in steam turbines and similar systems can be considerably reduced by:—

(*a*) The employment of suitable circulating pumps; in general,

the most suitable type is the rotary pump, which will give a continuous flow with a comparative absence of surging.

(*b*) The provision of suitable pipe work with gradual bends, etc.

(*c*) Preventing splashing and cascading of the oil.

(*d*) Removing unavoidably entrained air quickly; closely connected with this is the rate of circulation.

The authors lay stress on the design of the separating tanks, which should be accessible for cleaning and of such a shape as to facilitate deposition and drainage. It is pointed out that, in theory, the separation of water and the separation of solids should be effected at different temperatures, but that in practice, space and design do not always allow of this.

In his paper dealing with the practical problems of lubrication, Nicholson emphasizes, with regard to cylinder lubrication, that a small amount of oil of the correct type should be applied in the correct manner to obtain the best results. Overfeeding, especially in the presence of any solid impurities brought over by the steam, generally results in deposits in the cylinders and steam chests.

The author stresses the necessity of providing air compressors with a clean air supply and the correct quantity of oil, since impurities brought in by the air and excess oil exposed to the oxidizing effects of the heated air under compression are soon liable to cause deposits.

Nicholson advises that a plate indicating the correct lubricant should be attached to enclosed gear units by the makers, since they are able to obtain such information from the test bench.

As a result of observations over a length of time, Nicholson confirms the laboratory discovery that the addition of a small percentage of free fatty acid increases the effectiveness of a mineral oil when used for all erratic or partial lubrication, such as hand-lubricated bearings.

In discussing the lubrication of heated bearings, it is pointed out that the bearings should be designed to give a smooth flow of oil from all parts to the outlet, to prevent the accumulation of any deposits formed. It is essential that the circulation system should be driven independently of the machine so that circulation is obtained before the machine is started and after it is stopped.

DISCUSSION ON PAPERS IN GROUP II

14th October 1937

The CHAIRMAN, Mr. Charles Day, M.Sc.Tech. (*Past-President*), opened the meeting on Thursday morning, 14th October 1937, at 10.30 a.m. He welcomed visitors from overseas, and said that the group relating to internal combustion engines had been summarized by Mr. H. R. Ricardo, F.R.S., but as Mr. Ricardo was abroad, the summary would be presented by Mr. J. F. Alcock. The summary of the papers relating to reciprocating steam engines had been prepared by Mr. W. A. Stanier. The Chairman said that he had read the papers in Group II and considered that they formed a unique book of reference. He then called upon Mr. J. F. Alcock to read the report (p. 607) on the papers dealing with the lubrication of internal combustion engines, after which Mr. W. A. Stanier presented his report (p. 614) on the papers dealing with reciprocating steam engines.

On behalf of the meeting, the Chairman thanked Mr. Ricardo, Mr. Alcock, and Mr. Stanier for their most interesting summaries of the papers.

Discussion

Dr. H. S. HELE-SHAW, F.R.S. (*Past-President*), stated that he proposed to speak on only one aspect of the subject (which had been dealt with by more than one paper), namely, the reclamation of lubricating oil. In early days there was no attempt to save lubricating oil or even to economize it. He was old enough to have served as assistant engineer on a new tramp steamer, more than 60 years ago. There were no lubricators, except for the cylinders and oil cups on the main bearings and the oil was simply poured by hand from an oilcan on the shaft reversing gear and other parts, the piston rods being merely swabbed from a bucket of melted tallow. Surplus oil was lost in the bilge. When he (the speaker) delivered the Cantor Lectures on Friction before the Royal Society of Arts (1886), he gave an account of the work of a Belgian, M. Van den Kerchove, who was the first to investigate the subject of lubrication scientifically and, although he had proved what a great benefit scientific lubrication was, he never suggested that the lubricating oil should be recovered.

With the introduction of the internal combustion engine and the enclosed crank chamber a great economy was effected as the oil was

used over and over again, instead of being lost altogether, but it gradually became less efficient, and the time arrived when it had lost its fluidity to such an extent that it was thrown away, and was anything but a useful addition to the drains.

To-day, as could be seen from the latest books on lubrication, there were a number of processes of oil recovery and purification, chiefly by filtration and precipitation. In that way dirty oil could be purified and used again and again, for as was now generally recognized the oil itself did not wear out in use, and when purified it even improved with use.

The importance of saving oil would be realized when it was remembered (1) that up to the present mineral lubricating oil was chiefly imported, (2) that it was totally unlike rain, with which England was blessed abundantly, and which even when vaporized was never really destroyed. Oil once lost in the ground was lost for ever.

Mr. Beale, of Stream-Line Filters, Ltd., informed the speaker that as much as 97 per cent of the whole oil put in from the sump to a filter could be recovered in a perfectly good state for use. To prevent misunderstanding, that of course was only the oil left for filtration, as a certain proportion always disappeared in use, especially the lighter oil which was useless for lubrication, and although fresh oil was always required for replacement of losses there was still on the whole a considerable saving.

Mr. O. Thornycroft (Messrs. Ricardo and Company) said that reference had been made to the "frying" of lubricating oil in an internal combustion engine. Frying was a cooking progress in which the material was brought into contact with a hot surface. By far the hottest surface with which oil repeatedly came into contact in a high-speed engine was the under-side of the piston. In aircraft engines, that surface might reach a temperature as high as 350 deg. C., according to one of the papers, and, under the same conditions, the temperature of the cylinder surface would probably not exceed 200 deg. C. The continual increase in the power of the small high-speed engine, particularly the aircraft engine, rendered the question of piston cooling a very vital one. The designers of high-speed engines should consider whether the piston could be cooled by means of an oil jet, preferably issuing from the small end of the connecting rod. Preliminary experiments had been carried out on a supercharged single-cylinder engine of 5 inches bore and 5½ inches stroke, running at 2,500 r.p.m. An oil jet issued from the top end of the connecting rod, squirting 23 gallons of oil an hour against the under-side of the piston. Contrary to expectations, the oil consumption was not excessive, being no higher than 0·01 pint per h.p.-hr. Moreover, there was actually no measurable increase of

consumption when the oil jet was put into operation. By measuring the heat received by the scavenged oil, both with and without the jet, it appeared that the oil jet removed heat equivalent to 3 per cent of the total heat of the fuel consumed by the engine. The mean temperature rise of the oil issuing from the jet was found by calculation to be 90 deg. C. If it were assumed that 6 per cent of the total heat of the fuel was received by the piston (which was really a high figure) and that the heat received was the same with and without the jet cooling, then the mean temperature rise of the piston, above the cooling water temperature, must have been reduced by at least 50 per cent, due to the oil jet, and the maximum "frying" surface temperature reduced from approximately 350 deg. to something less than 230 deg. C. If such a lowering of piston temperatures could be achieved in practice, important reductions of oil sludging and ring-groove deposits would probably result.

Professor E. NORLIN (Statens Provningsanstalt, Stockholm) said that one factor affecting cylinder wear to a remarkable degree was the influence of dust or solids in the oil formed by wear or derived from the atmosphere. Tests made with one car and three lorries showed that, during a driving distance of 2,000 to 3,000 km. and employing used oil which had been passed through a centrifuge (the content of very fine solid matter still being about 0·22 per cent), the wear was greater than when the same distance was driven with fresh lubricating oil which was free from solid matter. With the private car, the wear was 70 per cent higher and with the three lorries, 30 to 40 per cent higher. Tests had also been made with oil containing 0·11 per cent colloidal graphite and oil without graphite, and greater wear was found with the graphitized oil. The ash content of the graphite was 1·6 per cent. Those tests showed that the solid mineral matter in the oil had a very great influence on the wear, so that emphasis should be laid on the necessity for using air filters and oil filters for the circulating oil and for replacing used oil by fresh oil after short periods.

Mr. J. CARTER (Glacier Metal Company) remarked that one of the major problems of the designer of compression-ignition engines was to provide a suitable lubrication system. To be able to compete with the petrol engine, there was a constant search for methods to improve the thermal efficiency of the compression-ignition engine. In order to do that, it had been necessary to strengthen components such as the crankcase, crankshaft, and connecting rod, thus allowing higher compression ratios to be employed. The strengthening of those parts, however, together with the increase in the rate at which the combustion

pressure rose, had, in turn, produced an increase in the mean combustion pressures, thereby causing high local stresses which tended to break down the oil film. In a petrol engine, the pressures on the bearings were at a maximum at one period only, i.e. when the engine was running at full throttle. Moreover, since at any other position of the throttle, the compression pressure and the combustion pressure were curtailed, then the heat generated in the bearing was less than at full throttle, and therefore the lubricant was assisted in that it had a smaller amount of heat to dissipate.

With a compression-ignition engine, however, the bearings often had to withstand a compression pressure as soon as the engine was started up, which in many cases exceeded the combustion pressures of a normal petrol engine. Therefore, the problem of adequately lubricating a compression-ignition engine was far more serious than that presented by a high-efficiency petrol engine.

He had carried out experiments with 2-cylinder and 4-cylinder compression-ignition engines in order to find the maximum load that the bearings would withstand. With the 4-cylinder engine, the connecting rod bearing load was increased to 3,770 lb. per sq. in. projected area, this giving a p/v load of 38,250 lb. per sq. in. projected area. The engine was run at an ungoverned speed of 3,000 r.p.m. While the engine was being run to find out how long the bearings would stand up under those conditions, the piston in No. 2 cylinder seized. When the engine was stripped, it was found that in three out of four cylinders the second ring down had seized under similar conditions. The injectors were on the side of the head remote from the thrust side of the piston, and it was at first thought that fuel was thereby being allowed to penetrate behind the rings, causing them to stick. In the 2-cylinder engine, however, the injectors were installed on the opposite side of the head, i.e. on the piston thrust side of the cylinder. Subsequently, that engine was run at a speed giving the same piston speed as that of the 4-cylinder engine. When the oil temperature was roughly the same as that at which the 4-cylinder engine had seized, the 2-cylinder engine also seized. All the rings were stuck in one piston, while in the other piston the second ring from the top had seized. It might be thought that an excess of fuel was being used, but the consumption was 0·406 pint per h.p.-hr. at the speed for maximum torque, giving a brake mean effective pressure of 102 lb. per sq. in. The frictional brake mean effective pressure recorded was 22·56 lb. per sq. in., the mechanical efficiency being 81·8 per cent. Those results appeared to contradict any suggestion that the fuel was passing by the rings. It was therefore concluded that the seizures were due to rupture of the oil film on the cylinder walls near the top dead centre. Consequently, a lubricating oil was needed which

would retain its viscosity in a compression-ignition engine running under conditions of continuous high stress.

The 2-cylinder engine had four main bearings and the oil pressures were measured at the point of greatest escapement, i.e. at the top of the bearing, in order to find the difference between engine gauge pressures and escapement oil pressures, so that some conclusion could be arrived at as to the feed pressure at the connecting rod. The outrigger bearing, which had no oil take-off and was supplied from the same main gallery as the other three bearings, showed 100 lb. per sq. in. with the oil at 100 deg. C. The other three bearings, when the outrigger was showing 100 lb., showed no pressure. A remark which appeared to agree with the suggestion made by Mr. Dicksee in his paper, and which required investigation, was that the feeding of the big end should be independent of the main bearings.

Bearing failures were chiefly due to flexing of the bearing in the connecting rod eye, mainly because the bearing shell pieces were made of softer material than the connecting rods. The hammering effect due to either crankpin or connecting rod cap distortion was the factor to consider in relation to bearing failures.

The scoring of the materials used for the big end, such as copper-lead, lead-bronze, and whitemetal, should also be considered because segregation of the lead allowed carbon compounds to form small points in the copper, which broke down the oil film and tended to raise the temperature on the shaft. Further, the embedding of the carbon compounds contained in the oil would have no effect on a whitemetal bearing, but with the harder bearing metals, there might be trouble, because they generated heat more quickly and they also resisted any carbon, and therefore more rapid wear must be expected.

Mr. F. Nixon (The Bristol Aeroplane Company) said that the problem of cylinder bore wear was fortunately one which did not affect aero-engines. His firm's practice was to employ nitrided steel cylinder barrels finished by honing; very high pressure scraper rings having a bearing pressure of about 70 lb. per sq. in.; and a so-called "high initial oil pressure" lubrication system. In that system, the pressure of oil entering the engine was about 230 lb. per sq. in. on starting, and it fell to the normal pressure of 80 lb. per sq. in. when an oil temperature of 35 deg. C. was attained. During that period there was also an additional jet of oil into the crankcase.

With regard to bearing temperatures it was extremely beneficial to reduce the oil temperature, but aero-engines presented difficulties in that the oil had to be cooled in a radiator, so that the higher the outlet oil temperature, the more efficient could be the design of the cooler.

Great improvements had been made in reducing the drag due to the cooler by recent developments with ducted coolers, in which an effort was made to regain some of the heat energy given to the cooling air by converting it into thrust. The improvements in coolers due to increased oil temperatures would, however, only be utilizable when new bearing materials capable of operating at higher temperatures, and oils suitable for those temperatures, had been developed. One of the problems of lubrication at actual operating temperatures was that of sludging, and oils more stable in that respect were urgently needed.

Referring to the use of hardened shafts with lead-bronze bushes, the speaker's firm was obtaining very successful results with reduction gear with fixed lead-bronze bushes on a soft shaft. By soft, a steel of a tensile strength of about 60 tons per sq. in. was meant.

The importance of correct breathing was shown by experiments in which the moisture in the breather gas was collected and found to be very strongly acid. The effects on the oil of tetra-ethyl lead in the fuel needed investigation.

The most urgent need was for an oil specification incorporating tests which would give results agreeing with those obtained in practice as, at present, the only reliable criterion of oil suitability was its behaviour in an engine. An oil combining the merits of extreme-pressure lubricants with suitability for operation in the rest of the engine would be very useful in avoiding complications due to duplication of the lubrication system, to increase the capacity of engine gearing.

Mr. M. ROEGIERS (Elektrion Oil Works, Ghent) said that several papers dealing with the performance of motor oils referred to the influence of the products of blow-by in the formation of deposits or sludge. The effect of that factor was considerable, and, so long as it was not accounted for in artificial ageing tests, those tests would continue to be discredited.

Some of the oils of the electro-chemically treated type, straight or blended, were now highly appreciated all over the world. That was due, not only to the excellent lubricating value resulting from their outstanding oiliness and high viscosity index, but also to the absence of sludge formation and the increased resistance to ring-sticking. Those oils, however, appeared to be unsatisfactory when examined by conventional laboratory ageing tests. What was the explanation of that contradiction between laboratory instability and practical stability? The oils in question had the outstanding ability of keeping in solution or taking into quasi-colloidal suspension most of the unburned oxidation products which were blown past the piston rings into the crankcase. None of the present laboratory ageing tests could show that.

In a state of solution or quasi-colloidal condition, the oxidation products were no longer harmful to the engine, and it might be assumed that the special physical and chemical character of those products further increased the "oiliness" of the fresh oil.

Modern mineral oils did not appear to possess those practical advantages, though modern refining methods had contributed much to the physico-chemical stability of the oils in the liquid phase. Such oils, however, still appeared to be unable to cope with carbonaceous products from blow-by. Modern mineral oils appeared to maintain an engine clean so long as the pistons, rings, grooves, and cylinders were very close-fitting, that was to say, so long as blow-by was negligible, but as operation was prolonged, the tightness diminished, and with such mineral oils, solid blow-by products were soon deposited. For those reasons, air-line operators in Europe had to reject many oils despite the fact that they might be classified as excellent by artificial ageing tests and quite satisfactory by engine-bench tests.

To support those assertions, he would summarize tests made during more than a year by the Belgian Airways Company *Sabena*, with ten Hornet engines, both new and completely overhauled. The oils compared were a modern refined mineral oil and an electrically treated oil. The specific gravity of the first was 0·885 as against 0·905 in the second; the colour was No. 2 as against No. 6; the viscosity at 100 deg. C. was about 20 centipoises in both cases; and the coke number after oxidation by the British Air Ministry method was 0·63 per cent for the mineral oil and 1·8 per cent for the electrically treated oil. The operating conditions for all the engines were mild, namely, 55 to 60 per cent of the rated power at cruising speed. Drainings were taken every 75 flying hours, and average samples were drawn from each oil draining and examined for deposit or sludge content by the centrifuge method of the American Society for Testing Materials (D.91–35). Some 60 drainings were tested and the sludge results provided the following indications: First, for new engines and for both oils, the first two or three drainings contained between 0·2 and 0·4 per cent of sludge. During those periods, the low oil consumption indicated that the amount of blow-by was very small. Second, over longer periods of engine operation, the sludge content of the mineral oil increased fairly regularly up to about 2 per cent. There were occasional peak values of 2·5 per cent and even 3·2 per cent. Those peak values appeared to be the result of occasional ring-sticking. With the electrically treated oil, the amount of sludge remained fairly uniform at about 0·4 per cent, even with engines which had been operating for over 800 flying hours without overhaul. Occasional peak values never exceeded 0·8 per cent. On being stripped, the engines, some of which had totalled over 800 flying hours,

always showed much more sludge deposit in the hollow crankpin with the mineral oil than with the electrically treated oil. General cleanliness and wear on rubbing surfaces were comparable for the two oils, except for those parts where boundary lubrication was prominent (e.g. gudgeon pins), where the electrically treated oil was better.

That mineral oils of the quality described failed to complete the test on the very high-output engines now being built in the United States, must not be attributed to lack of chemical stability, but it strengthened the belief that those oils were deficient in "oiliness".

The special solvent properties of the electrically treated oils had proved to be of still more importance for compression-ignition engines. Mineral oils, either conventionally refined mineral oils or modern refined mineral oils which had been blended with electro-chemically made concentrated oil, dissolved or took into almost colloidal suspension soot and most of the other blow-by products which were often so abundant in Diesel type engines. That feature made possible much longer operating periods between drainings.

Dr. E. W. J. Mardles (Royal Aircraft Establishment, Farnborough) said that attention had been drawn by authors and speakers to the very important facts that the impurities which found their way into an oil and the combustion products from the engine seeemed to accelerate oxidation of the oil, that the deterioration of oils in service was obviously not covered by the usual oxidation test, and that laboratory ageing could not be used alone to judge the quality of a motor engine oil. Mr. Barton had pointed out (p. 407) that the results from the various oxidation tests differed. Having compared the Indiana oxidation test with the Air Ministry blowing test, he found that the results with the same oil disagreed, but that was not surprising, because the Indiana test was based on the sludging tendency of oils, whereas the Air Ministry oxidation test dealt with increases in viscosity. It was known that there was no well-defined correlation between sludging and the viscosity rise due to polymerization during oxidation. Better agreement between the two tests could be obtained if they took into consideration both viscosity rise and sludging tendency.

Some very important facts must be considered in any oxidation test of oils. First, the rate of deterioration of an oil was not proportional to time, so that if two oils, A and B, were taken, and a graph was plotted relating time and oxidation change, two different types of curve would be obtained. In that particular case, oil A at first was worse than oil B, then the curves crossed, so that there was a time when they were similar, and then ultimately the worse became the better. Second, the deterioration of oils must not be measured by any one particular change.

All the changes—acidity rise, viscosity rise, sludging, tendency to emulsification and so on—must be considered. Third, and most important, the temperature coefficient of oils differed very considerably. One could take two oils and plot the rate of deterioration by oxygen absorption or of sludging or of viscosity rise, or of any other change at different temperatures, and get two distinct curves. A vegetable oil or a blended oil seemed to have a higher oxidation rate at lower temperatures, but at higher temperatures it behaved better than the straight mineral oil. The temperature coefficient was also affected by impurities, by inhibitors, etc. In an oxidation test, at least two times and two temperatures should be considered. Some of the authors seemed to think that the results of an oxidation test were a snare and a delusion, but it was possible, in the light of the data now available, to find some sensible solution for the problem of finding suitable laboratory tests and to tell the engineer the probable behaviour of any particular oil in the engine from the laboratory oxidation results.

Professor H. A. Everett (Department of Mechanical Engineering, Pennsylvania State College) said that, in his summary, Mr. Alcock had pointed out that the paper by Commander Yeates showed a slight difference in the wear occurring in the starting or in the early operation of the engine from that shown in the paper contributed by Mr. Keller and himself. The tests which they had reported were run on an 8- or 9-hour basis, and, taking the graph in Fig. 1 of Commander Yeates's paper (p. 596), the number of hours' run per start would come in the first of the co-ordinate group, and the wear would be about 4/1,000 inch per 1,000 hours. Their engine tests corresponded to about 60,000 miles' running. That would give about 1/1,000 inch cylinder liner wear from 15,000 miles, which was the figure given in their paper.

A comparison was frequently made between theory and practice, often to the detriment of theory. Theory, however, was simply an explanation. If theory and practice failed to agree, it was the fault either of the theory or of the manner in which the practical results were reported. The theory might be incorrect and the explanation bad, or the results of the practical experiments might be reported in such a way that they did not give the necessary data. In either case, there was a differentiation, and frequently the practical man damned the theorist and the theorist accused the practical man of not being familiar with the subject. The question of lubrication, however, was so complex and presented so many variables that any theory had to be a simplification, and that simplication must be borne in mind. Thus, viscosity was probably the most important single property which had to be dealt with in lubrication problems. The theorist was practically forced into a

consideration of viscosity as a constant, as otherwise one got into very complex situations, but viscosity was far from being constant. Professor Bradford had directed attention (p. 23) to some work recently done at the Pennsylvania State College on the effect of pressure on viscosity. The data on viscosity were given up to about 30,000 lb. pressure. For a given temperature, the difference in absolute viscosity was at least 3 to 1 and in some cases still higher. That difference in viscosity was due to pressure only, so that, if the basic assumption was that viscosity remained constant, some difference must be expected between theoretical and practical results. Conversely, the practical results as given were in many instances not what they seemed. A man might honestly carry out experiments in which he thought he had kept one variable constant, but he might not have done so. He might run two oils in a bearing and find some difference; he might attribute it to the oils but it might be due to something else. Until one knew that one was getting the effect of one variable only, practical results were usually of entirely secondary importance.

It had sometimes been thought that wear indicated the character of the lubrication. If there was perfect lubrication, there was no wear, whilst if there was wear, there was obviously not perfect lubrication. Therefore, when comparisons were made in which one wished to investigate lubrication, one often needed to be guided by some sensitive method of indicating wear, and it was really with the idea that such a method might be useful, that Mr. Keller and he had described the iron contamination method for indicating wear. If that method made it possible to determine the amount of wear which took place in an engine operating at 60 m.p.h. in 10 minutes, it would correspond to the increase of wear observable for 10 miles, and that was remarkably sensitive. With an instrument of such sensitivity, great care was required, but it was very useful.

Investigators were apt sometimes to cover up unknown factors by some new term. Recently there had been much discussion about oiliness. He did not wish to suggest that such a thing as oiliness did not exist, but he had come to the conclusion that, in many instances in which oiliness had been charged with certain differences, particularly in fluid film conditions, they might really have been due to changes in viscosity. It might be that, through considering viscosity constant when it was not, people had charged the vagaries they had found to oiliness when they were not really due to oiliness.

Mr. A. Taub (Vauxhall Motors, Luton) observed that nobody disagreed with the corrosion theory. Certainly he did not; he knew the evils of corrosion. There was a distinct and outstanding difference

between wear in America to-day and wear as it existed in this country. It was one thing to work on one or two engines in a laboratory, but it was quite a different matter to have to find a practical remedy. Of course, corrosion was important. In practice, it should be dealt with by allowing the engine to warm-up fairly quickly. It had often been proved that, when sufficient oil was provided, the evils of corrosion were reduced, but one must not imagine that corrosion was only a factor during cold starting. Thus, a motor car that had been kept overnight in a warm garage could be started fairly quickly and kept running, perhaps being started twice in 400 miles, and yet there was bore wear. Corrosion must play a part there, because the ratio of bore wear with 30 deg. difference of metal temperature, caused by the use of alcohol or water, was 3:1. A fact which suggested that the corrosion theory must not be held to cover the entire issue of bore wear was that a difference of 2:1 in the mixture ratio, i.e. a 12:1 mixture ratio as compared with a 14:1 mixture ratio, made a difference of 5:1 in the bore wear. That was not under starting conditions, but under constant operating conditions, with a car out of a warm garage and with fairly continuous running. The remedy was to make engines that would operate on a better mixture ratio.

He had been charged with being in favour of high tension rings, but he was advocating, not high tension rings, but keeping the blow-by and oil from going past the rings. If lower tension rings could be made to serve the purpose, he would advocate them, and he was going to try them.

There was some misunderstanding about tapered face compression rings. The tapered faces were only temporary. When tin plating was first applied to cast iron pistons, it was found that the initial consumption of oil was bad. Therefore, a slight taper was put on the face, so that the compression rings would control the oil for at least 1,500 miles, until the oil ring was bedded in, and after 1,500 miles there was no taper.

The wide ring had been advocated and, if a wide ring could be made that would not give serious blow-by, it would be a wonderful invention, but the rings chattered, and when there was chattering, then blow-by occurred. When the rings were widened, the break in the blow-by occurred too early. A normal engine would have a fairly uniform blow-by of from $\frac{1}{2}$ to $\frac{1}{4}$ cu. ft., but when it came to the breaking point, it would go up to 6 cu. ft.

Copper-lead bearings were not impervious to acid in the oil, and one of the reasons why he was so anxious to lessen blow-by was that he wanted to keep the acid out of the oil.

Ventilation was an important matter. Assuming that soft rings were

good and that blow-by could be kept at $\frac{1}{2}$ cu. ft., it was still necessary to get rid of the results of that blow-by, so that crankcase ventilation was necessary. When thermostats and ventilators were fitted in the United States, machinery and equipment for re-boring cylinder blocks had practically disappeared, and it was now the work of a specialist.

Monsieur H. Brillié (Former Consulting Engineer to the Compagnie Générale Transatlantique), referring to the paper by Professor H. Nordmann and Herr J. Robrade, said he had recently made observations on the lubrication of locomotive axle bearings where the perfect functioning of such bearings with lubricating pads presented certain difficulties. One of the chief difficulties resulted because the bearings were machined by the manufacturer, whereas the client wished to be able to use such bearings immediately they were assembled, on full load and at maximum speed, without any previous running-in. After the adoption of special grooves, which only partially solved the problem, he had made comparative tests without any precise measurements of quantity but with careful observation of the state of the surfaces after the tests. The results of the tests, which were carried out first with an oil bath, were as follows: With an oil of average viscosity (about 2 poises at 35 deg. C.), the results were very nearly satisfactory, but left room for improvement. With a more viscous oil (about 7 poises at 35 deg. C.), there was no appreciable change. An oil with a viscosity of about 11 poises at 35 deg. C., to which 2 per cent of "Oildag" colloidal graphite had been added, gave satisfactory results. The oil of average viscosity, after 2 per cent of "Oildag" had been added, also gave improved results, and was considered satisfactory. The tests were carried out without any previous running-in under the most severe working conditions, lubrication being effected by an oil bath. During operation, lubrication was obtained by wick ascension of about 7 to 8 cm., the oil level being below the lower lines of the bearing. The question arose whether the colloidal graphite would pass the wicks. Under the microscope, the oil containing "Oildag" was found to be unchanged after passage through the wicks. The bearings were then wick-lubricated with oil having a viscosity of 2 poises, to which 2 per cent of "Oildag" had been added. The results were entirely satisfactory, and entirely comparable with those obtained with the oil bath. The difference between the results obtained by the speaker and those given by Professor Nordmann and Herr Robrade might be attributed to two causes: First, the nature and arrangement of the wicks employed. The pads which the speaker used were installed so as to favour capillary action and ensure abundant lubrication. Second, to ensure the passage of colloidal graphite through wicks it was important to employ

colloidal graphite which had the smallest and most uniform particle size as well as the greatest purity. The tests were being continued.

Professor D. Dresden (Utrecht, Netherlands) expressed his admiration for the masterly way in which the General Reporters had given a review of the mass of data contained in the papers. The premises were not sufficient, but Mr. Alcock had managed to draw conclusions from them. Mr. Alcock had also shown the relative importance of statistical data. Even correlations did not always give an explanation.

He could confirm Mr. Nixon's statement that lead-bronze bearings could be used successfully with soft shafts. Designers in Holland made use of that combination where the steel had a tensile strength of about 60 tons per sq. in., which was the figure Mr. Nixon had given. He could also confirm the statement that the tendency to form sludge was really increased by adding new oil; at least, that was what he had found in turbine practice for about eight years; that experience, however, was not recent.

Mr. A. Wolf (Department of Oil Technology, Imperial College of Science and Technology) said that the two most striking general improvements which had been made in lubricating oils during the last ten years or so were the following :—

(1) The very great increase in the chemical stability of turbine and internal combustion engine oils as the result of the introduction of solvent refining processes. Mineral oils which had been treated in that manner were remarkably resistant to sludging even under severe oxidizing conditions, and had much flatter viscosity-temperature curves than when refined by the older conventional methods.

(2) The improvement in the lubricating power of oils under severe conditions by the addition of small proportions of certain substances which might be termed "adjunct lubricants" and, in a sense, regarded as the "concentrated essence of oiliness". Those additions increased the "oiliness" or adhesiveness of the oil, and enabled it to give reasonably satisfactory lubrication even when the supply was very sparing and pressures and temperatures were relatively high.

The speaker knew of individual engines and machines which, owing to lack of alignment, faulty bearing design, incorrect methods of supplying oil to the frictional surfaces, etc., could not be operated at all without the addition of an appropriate adjunct lubricant to the normal lubricating oil.

Closely allied to the second improvement (and to a considerable extent, responsible for forcibly reminding the lubrication technologist that viscosity alone was insufficient to provide lubrication) was the

lubrication of hypoid and other transmission gears operating under severe conditions.

The typical extreme-pressure lubricant which had been developed for the lubrication of such gears consisted of a viscous mineral oil which served as a vehicle and diluent for some "active" substance which would impart to the oil the required load-carrying capacity or film-strength. Film-strength, although not identical with adhesion and oiliness, was probably closely related thereto, and here again one had an example of the lubricating power of an oil being greatly increased by the addition of a comparatively small proportion of some "active" component.

Mr. P. Jackson (Messrs. Mirrlees, Bickerton and Day, Ltd.) observed that in compression-ignition engines operating up to 90 lb. per sq. in. brake mean effective pressure, he had not found different lubricating oils to vary much in their effect on the sticking of rings. The construction and design of the engine had far more influence than any change of lubricating oil on that particular aspect. His experience was that the clearance of the top ring in particular had the greatest influence. With a clearance of less than 0·006 inch on a piston of 6 inches and over, the top ring would stick with any type of lubricating oil, but when there was a clearance of 0·008 or 0·010 inch the top ring would not stick. That was his general conclusion, but there had been occasional exceptions.

Again, the amount of cooling of the cylinder walls considerably influenced the sticking of the piston rings. Where the cylinder water jacket was carried down to the centre line of the gudgeon pin at the bottom of the stroke the rings were relatively free from sticking, but on a similar design of engine where the length of the water jacket was curtailed and did not extend below the ring band at the bottom of the stroke, sticking of the piston rings would be a difficulty.

In the development of a high-speed, two-cycle engine operating at 1,800 r.p.m., it was at first impossible to get the engine to run for more than 3 or 4 hours without the rings being stuck, but ultimately periods up to 100 hours had been made possible by three changes: First, by giving the piston rings more clearance; second, by giving more cooling to the cylinder walls; and third, by using an oil-cooled piston. The two last-named steps both led to cooler pistons and cooler operation of the rings.

A lubricating oil containing ethyl lead had been of considerable advantage, but since such an oil was not normally marketed, it was entirely a laboratory condition.

Regarding the influence of combustion on sludge formation, he had

found that with an engine of open chamber design with a definite degree of swirl giving clean combustion when using an ordinary fuel of 75 seconds viscosity, there was no trouble with ring sticking or sludge formation, but when a heavy oil of 200 seconds viscosity was used, again with clean combustion and an increase of fuel consumption of no more than 5 per cent, ring sticking and sludging of the lubricating oil were both encountered. The reason for that was not clear.

Solvent-extracted oils and electrically treated oils had little influence on ring sticking, but greatly influenced the internal cleanliness of the engine. With such oils there was much less sludge in the crankcase and the general appearance of the running parts was far cleaner.

The carbon formed differed considerably in its properties according to the brand of lubricating oil used. With some lubricating oils the carbon was soft and the rings could be relatively easily withdrawn, whereas with other oils the carbon was hard and piston rings had to be broken out of their grooves.

Another point which influenced the lubrication of an internal combustion engine was the size of the scavenge oil pump on dry sump lubricating systems. It was generally considered that the duty of the scavenge pump was to draw the oil out of the crankcase and deliver it to the oil tank and that the bigger it was for this purpose the better. He found, however, that an excessively big pump drew a mixture of oil and air and that the consequent aeration of the lubricating oil increased frothing, particularly where there was any slight dilution by water and that under such conditions there was excessive emulsification and sludge formation.

He was interested to read the remarks of Mr. Ottaway on the effect of the type of combustion chamber on lubricating oil consumption and cylinder wear. He agreed with Mr. Ottaway that the open type of chamber gave in general a lower lubricating oil consumption and a lower degree of cylinder wear than the ante-chamber type of combustion chamber. In two instances in his experience an engine was changed from open chamber to ante-chamber type and in both cases the fuel consumption and the combustion were better, but the lubricating oil consumption and the degree of cylinder wear were considerably increased.

Mr. E. A. Evans (Messrs. C. C. Wakefield and Company) said that Professor Dresden had given him the impression (he was sure, unintentionally) that there was a definite danger in adding new oil to old. Formerly, it was bad practice to add new transformer oil to old, and then it was thought to be bad practice to add new turbine oil to old. It was a bad practice in many cases, because the new oil, drawn from

another field, did precipitate the partly dissolved sludge, but it was wrong to think that it was bad practice to add new oil to engine oil actually in service. It was perfectly safe to add new oil to engine oils.

Mr. Nixon had asked for specifications for engine oils, and Dr. Mardles had explained some of the difficulties in that connexion. There were very real difficulties in the way of preparing specifications for oils, and the discussion had brought out many of those difficulties. Mr. Roegiers, for example, had mentioned the difficulty in obtaining correlation between laboratory tests and road tests.

Mr. Nixon had asked when it would be possible to get rid of sludge. He could not answer that question, but a very great endeavour was being made to reduce sludge. He would like to utter a warning about the so-called growing popularity of inhibitors. He thought that very few inhibitors were actually in use. The difficulty was that inhibitors, at any rate organic inhibitors, would not withstand a high temperature. Inhibitors were used for turbine oils, but they were added mainly to keep the acid low. They did not, so far as he knew, reduce the sludge very markedly. It was time to pause and consider the question of additions to oils. Some people put materials into oils to increase the film rupture strength. He did not condemn that, but caution was required. Additive compounds to increase film rupture strength were used for gears, but it was necessary to proceed cautiously when it was a question of engine lubrication.

Mr. H. J. Young (Sheepbridge Stokes Centrifugal Castings Company) remarked with reference to the paper by Mr. Pearce that there was a relation between wear and toughness and machineability; for example, when a cast iron of low Brinell hardness number (say 200) machined less easily than another iron of, say, 240 hardness number, it wore better. The surface of those tough irons retained oil film in position; hardening to be efficacious must not detract from that property. Mr. Pearce described the occurrence of sulphur in iron as "occasional crystals of manganese sulphide", but an enlarged sulphur print, as published by the speaker in 1921, did not confirm that description; moreover, the sulphur content profoundly affected the wearing property. That increased silicon reduced wear-resistance was a testimonial to Lanz Perlit iron. On the other hand, the speaker also obtained good wearing properties from irons much higher in silicon and believed that that end of the scale was valuable. Owing to its tendency to segregate, the speaker's experience of phosphorus had been that large castings could not carry with safety as much phosphorus as smaller and thinner ones. That pearlitic iron was more resistant to wear, as

enunciated by Diefenthaler, was accepted universally, and in 1934 the speaker demonstrated that the finer the pearlite lamellæ, the better wearing the iron.

With reference to the paper by Mr. Taub, the speaker pointed out that the service conditions of the six engines giving average bore wear were not stated, nor even whether those engines were fitted with high-tension rings in accordance with the author's present theories. Obviously, the result was either a "special" one or settled once and for all time the problem of cylinder wear. That hammering of piston rings weakened them locally, seemed to the speaker to depend upon the section of the rings concerned. The paper dwelt upon the disastrous effects of blow-by and ring chatter, and suggested that they were maladies common to English engines, but the speaker believed few engineers in Great Britain would agree. A high-tension ring, coupled with an increased supply of a thinner oil, appeared to have much to commend it, but the fact that six engines in America gave many times less bore wear than the best commercial automobiles running in this country, pointed to the lack of necessity for doing anything more than was done on those engines, but what that was, the author did not say.

Communications

Mr. J. F. Alcock (Ricardo and Company, Ltd.), in a written communication, amplified the comments made in the report upon Commander Yeates's wear figures. It was there stated that, if Fig. 1 (p. 596) of Commander Yeates's paper were plotted with the x scale inverted, it showed a roughly linear relation between the frequency of starting and the rate of wear. Fig. 1 made that clear. One of Commander Yeates's wear figures, that at about 2 hours run per start (500 starts per 1,000 hours) had been omitted, since it could not be included without cramping the rest of the scale. In any case, the small scale of the diagram in the paper made it impossible to locate that point with accuracy. It appeared, however, from that point and from other data that the wear per start diminished when the running period was very short. Thus, for two-hour runs, extrapolation of the line gave a wear of about 0·012 inch per 1,000 hours, which was rather on the high side. That suggested that the damage caused by each start was not completed until about five hours after the start; possibly the run-in surface was roughened either by corrosion or by abrasion due to lack of oil, and took about 5 hours to re-establish itself; if, during those 5 hours a fresh start was made, less damage was caused, since there was a less perfect surface to injure.

Another interesting point was the kink in the line at about 10 starts

per 1,000 hours (100 hours run per start). It was difficult to explain if all conditions were comparable, but it might be that most of the engines with long running periods were marine engines, which would have cleaner air than land engines and thus would have less abrasion wear. Perhaps Commander Yeates could give further information.

The crosses on the diagram referred to wear figures from two Brotherhood-Ricardo sleeve-valve oil-engines, one running for a week at a time non-stop, the other for eight hours a day, with a short lunch-time stop which was not counted as the engine did not become cold in the interval. In both cases the sleeve was of annealed crucible cast-iron

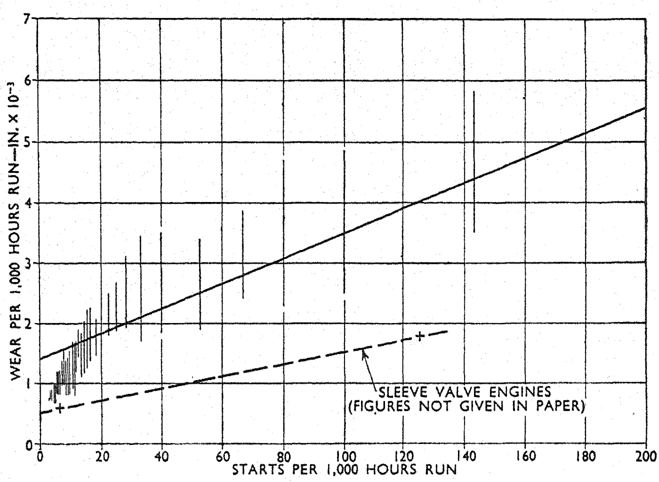


Fig. 1. Data from Fig. 1 of the Paper by Commander Yeates

Plotted with inverted x scale.

(not centrifugally cast) of the following percentage composition: total carbon 2·85, silicon 2·20, manganese 1·0, sulphur 0·055, phosphorus 0·23, chromium 0·40, and molybdenum 0·40. The pistons were of cast iron. In each case, the running period was about 10,000 hours. The ratio between running and starting wear was about the same as in Commander Yeates's engines, but the actual wear was about half his mean figure. That was probably due, in part, to the quick heating-up of the sleeve, which had an oil film between it and the cylinder, and, in part, to the improvement in oil distribution caused by the circumferential sleeve motion.

In drawing attention to the relationship between wear and starting frequency, Commander Yeates had brought out a very useful line of

investigation, and it was hoped that others with similar data would also produce them. Such data would be particularly valuable for small engines, since Commander Yeates's figures related to engines of 12-inch bore and over. With small engines, too, there was probably a better chance of picking out cases of identical engines with different running periods.

Mr. C. H. Barton (Asiatic Petroleum Company), referring to Dr. Mardles's remarks concerning the comparison of the results of Indiana and Air Ministry oxidation test results in his paper (p. 407), pointed out that the results were merely used to indicate, in a general way, the relative resistance of the four oils to oxidation. The comparison was made primarily between the Indiana "sludge" figures on the one hand and the Air Ministry coke number increases on the other. Apart from the indications which they gave of oxidation stability, neither of the tests was a reliable guide to sludging tendency.

In most aero and automobile petrol engines thickening of the lubricating oil in use was not sufficiently pronounced to give rise to comment, even when the effect of fuel diluent was allowed for. In Table 1 of his (Mr. Barton's) paper the increase in viscosity (expressed as "viscosity ratio") of the four oils in the Air Ministry test placed oils A, B, and D equal and all superior to oil C. The corresponding results for the Indiana test were not quoted in the Table, but they were available and showed that, after oxidation for 50 hours, the oils fell in the order B, D, A, C for "viscosity ratio" at 100 deg. F.

Mr. H. N. Bassett (Cairo, Egypt) wrote that the statement by Mr. Chatel (p. 439) that the lubrication of whitemetal piston rod packings could be neglected safely, was only true where oil was supplied to the cylinders in the form of mist. Otherwise, oil must be fed direct. Where the packing consisted of whitemetal rings and bronze rings, ample lubrication was essential, and its absence would lead to excessive wear of the packing.

Referring to the paper by Messrs. Dolton and Mackegg, used Diesel engine oil might contain a proportion of sludge which was soluble in the oil when hot (say 180 deg. F.) and should be removed. If the oil was taken from the sump at a lower temperature and then heated to 180 deg. F. before being passed through the machine, the sludge was not removed but went into solution, only to be thrown down again when the oil cooled. It was better, therefore, to allow the oil to settle and to draw off from the top to the heater. Otherwise, the cooler would become choked with deposited sludge. That observation applied with equal force to oil used in turbines.

Messrs. Fogg and Jakeman stated that the water content of an oil should be kept below 0·02 per cent. Inasmuch as that percentage made no difference to the appearance of the oil (it might cause cloudiness if the oil were cooled to 0 deg. C.) it would be interesting to know how that extremely small quantity of water was to be measured. Possibly a measurement of the dielectric strength would assist, though probably not quantitatively. The conclusions reached were particularly interesting in view of Mr. Samuelson's work (p. 269) on emulsions of water and oil for lubricating journal bearings.

The flashpoint quoted by Mr. Freeman for Diesel engine air compressor oil in his paper (p. 478) was higher than required, 360 deg. F. being quite high enough. The viscosity of the oil was rather high, and might well be within the following range: 300–350 sec. Redwood at 100 deg. F., 110–125 sec. Redwood at 140 deg. F., and 50 sec. Redwood (minimum) at 200 deg. F. An oil of high flashpoint could be made only from heavy stocks, which tended to gum under running conditions in air compressors. Such deposits often prevented the valves from seating properly, so that wire-drawing of the air took place, with continual rise in temperature until ultimately the deposit might become incandescent, and cause an explosion. The remedy was to prevent the formation of the deposits by using an oil of relatively low viscosity which would lubricate as effectively as a much more viscous oil without forming deposits.

The statement (p. 476) that oils from different suppliers could be mixed, provided that they had the same general characteristics, was only true when the oils were made from similar crudes. Otherwise, mixing might lead to the precipitation of oxidized compounds normally held in solution at the running temperature of the oil.

The asphalt content of used oils was not necessarily due only to incomplete fuel combustion as stated by Herr Kadmer (p. 502). Experiments made in the writer's laboratory with an oil initially free from asphalt showed that, after prolonged heating at 105 deg. C. (14 days and nights), considerable amounts of bodies coming within the Institution of Petroleum Technologists' definition of hard asphalt were formed. Incomplete combustion led to the formation of gummy deposits like lacquers * which were different from the asphaltic compounds frequently found in used motor oils.

The writer would like to ask Mr. Stanier whether any difficulty had been experienced with the formation of deposits in the annular space round the periphery of the piston valve liners. The groove was presumably comparatively narrow and shallow and provided with holes

* Bouman. Proc. World Petroleum Congress, 1933, vol. 2, p. 248.

through which the oil-steam mixture entered the piston valves. Steam frequently contained boiler salts which assisted in the formation of deposits, especially with compounded oils containing a proportion of fatty oil. If deposits formed in the groove, the whole value of the method of supplying the lubricant to the piston valves was lost.

The independence of wear-resistance and hardness referred to by Mr. Young (p. 602) had been shown by Hudson * in his work on wear in polishing plated surfaces. Except in the broad sense that the hardest surface (platinum-plated) showed the least wear and the softest (silver-plated) showed the most, the hardness number of the material was no guide to wear resistance under polishing conditions. Conditions in a cylinder which had been properly run-in were polishing conditions, and hence wear could be expected to be independent of the hardness of the wall. Brownsdon (1936 †) found in his work on metallic wear under lubricated conditions that, in a series of alloys such as the brasses, wear might be influenced by the composition, though it was independent of the hardness. Sawin's work (1937 ‡) supported the belief of the independence of wear and chemical composition, but on the other hand, it was well known that certain cast iron mixtures resisted far better than others when used in cylinders, the hardnesses being approximately the same. Jannin had shown that the principal cause of wear in bearings was the lack of finish of the shaft, and it was not unreasonable to suppose that surface finish in cylinders and rings was connected with wear resistance.

The scoring during moments of dry contact was due to molecular adhesion, and the consequent tearing-off of portions of the surface. Scoring might induce skin fatigue, but it was doubtful whether skin fatigue was the prime cause of wear.

Mr. A. Beale (Stream-Line Filters, Ltd.) wrote that, in his paper, he mentioned that there were many instances of engines in which the oil had not been changed in ten years. In the summary, Mr. Alcock had suggested that, in such cases, not much of the old oil would be left owing to the continued addition of make-up oil. That was true under ordinary conditions, but it was of interest that, in a number of cases, engines had run for prolonged periods without the use of any new oil, make-up being supplied by the use of filtered oil from other, similar engines. In addition to such prolonged experience with Diesel engines, the writer had had similar cases with bus and aero-engines.

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* Jl. Inst. Metals, 1933, vol. 52, p. 101.
† Jl. Inst. Metals, 1936, vol. 58, p. 15.
‡ *Machinery*, 1937, vol. 50, p. 165.

Mr. C. A. Bouman (Delft Testing Station, N. V. de Bataafsche Petroleum Mij.) wrote that, according to various authors of papers in Group II, laboratory tests in general did not give reliable information on the behaviour of the oil in the engine. The various chemical processes (oxidation, polymerization, carbonization, etc.) to which the lubricating oil was exposed in internal combustion engines were very complex. Moreover, their character varied with the local engine temperatures. Especially in respect of carbonization of the lubricating oil on the hot walls of the combustion space, in the ring grooves, inside the piston, and on the piston crown, results from actual engine tests showed important discrepancies when compared with the results from laboratory oxidation tests, "baking tests", "sticking tests" and the like.

On the other hand, the deterioration of crankcase oil (as measured, e.g. by the percentage of asphaltenes in the crankcase oil) might show a fair correlation with several laboratory oxidation tests at moderate temperatures. But, as in most engines the contamination of the lubricating oil was mainly due to soot formed by incomplete combustion of the fuel, it was clear that laboratory oxidation tests could not predict to what degree crankcase oil would be contaminated in practice. That was especially true for Diesel engines, which were more liable to sooty combustion than gasoline engines.

The conclusions were that:—

(1) As regards the deterioration and carbonization of lubricating oil at high temperatures in the combustion space, in the piston ring grooves, etc., laboratory oxidation tests could not be used to predict the behaviour of the oil, because of the very complicated nature of the processes involved.

(2) As regards the contamination of the lubricating oil in the crankcase, laboratory oxidation tests could not be used to predict the behaviour of the oil because the contamination products mostly originated from sooty combustion (of fuel *and* lubricating oil particles) in the combustion space.

The writer believed that the degree of contamination of the crankcase oil by soot could, in some cases, be predicted more accurately in the following way: Calling G the weight in kilogrammes of lubricating oil in the sump and in the oil-circulating system, V the lubricating oil consumption in kilogrammes per hour, a the quantity of soot entering the sump oil in kilogrammes per hour, x the soot content of the lubricating oil at the time t, then, under certain conditions * the change in soot

* Bouman, C. A., *The Oil Engine*, 1937, April, p. 362, "Limitations of the Amount of Soot in the Crankcase Oil of Diesel Engines".

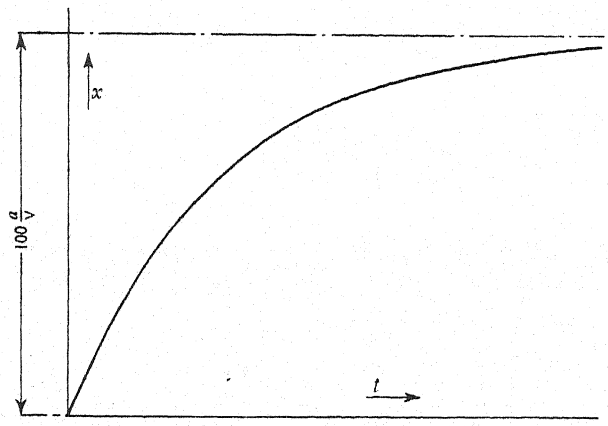

Fig. 1. Percentage of Soot in Crankcase Oil

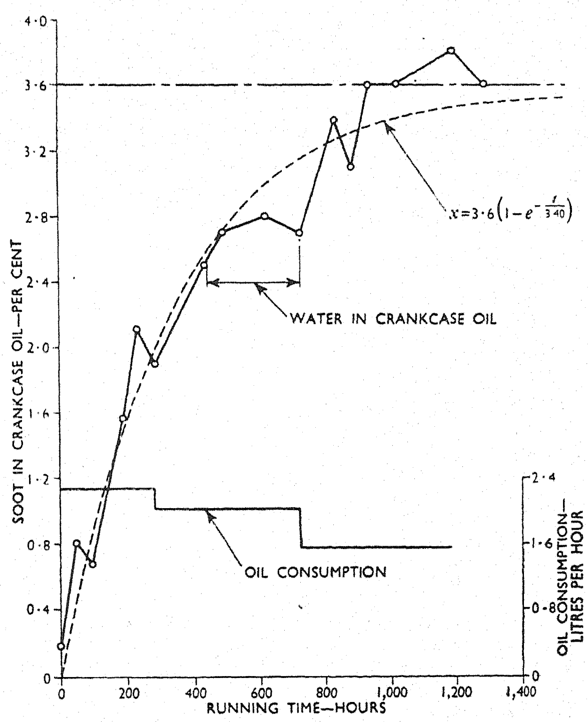

Fig. 2. Contamination by Soot of the Crankcase Oil of a 640 h.p. Diesel Engine

content with time would be given by the equation (represented graphically in Fig. 1):—

$$x=\frac{100a}{V}\left(1-e^{-\frac{V}{G}t}\right)$$

A practical example concerning a 640 h.p. Diesel engine at a pumping station running at constant, nearly full load was given in Fig. 2. For that case, the following values were found: G=660 litres=600 kg.; V=1·95 litres per hr. or 1·77 kg. per hr.; a=0·064 kg. per hr. The explanation of the final deviation from the theoretical curve in Fig. 2 was that towards the end of the 1,200 hr. running period the oil consumption had decreased to some extent due to thickening of the crankcase oil.

Mr. A. G. CAHILL (Messrs. Hopkinsons, Ltd.) wrote that Mr. A. W. Empson gave certain figures showing the results obtained by treating Diesel lubricating oil with "ordinary centrifuges" in marine service, which he had classed as "typical". It could only be assumed that the centrifuge plant was too small, or incorrectly operated, as the figures showed the oil to be in very poor condition indeed. Surely water percentages of 2·9 and 8·00 were "extraordinary" with an efficient purifier? Actual figures reported by one of the largest oil suppliers for oil which had been in continuous service for eight months in a Diesel-engined tanker were appended, and it was interesting to compare them with the "typical" figures in Mr. Empson's table. The centrifugal purifiers installed were of the "ordinary" type, and no special treatment was given to the oil except a "water wash" added to the feed.

Specific gravity	0·920
Saybolt viscosity at 140 deg. F., sec.	108
Saybolt viscosity at 210 deg. F., sec.	52
Acid value as SO_3, per cent	0·004
Moisture	Nil.
Sludge	Nil.
Impurities	Trace
Fuel dilution	Nil.

Mr. A. W. EMPSON (London) wrote that the installation of centrifugal separators for the purification of lubricating oil was now common practice in all modern power plants, and considerable economies were effected by their use. The results with internal combustion engines, however, left much to be desired, as the steady accumulation of colloidal impurities such as carbon, etc., which would not be eliminated by ordinary centrifugal treatment, ultimately involved entire renewal of the oil. Various methods devised in order to enable those impurities to be dealt with in a centrifuge included coagulation by means

of chemicals in order to produce large clots, or enmeshing them in activated earths prior to centrifuging. Those processes involved preliminary settlement, and hence could not be worked continuously, and furthermore, the yield of treated oil was often as low as 60 per cent.

Investigations by the writer had led to the discovery that plain water could be used as a coagulant with much greater efficiency, and plant had been evolved for carrying out the process on a continuous basis as opposed to the batch treatment necessary with other methods. The process depended upon the fact that colloidal carbon in a hydrocarbon oil possessed a negative electrical charge, and so long as that charge persisted, the particles were subject to Brownian movement. Water atomized in such oil assumed a positive charge, with the result that the charges on the carbon particles were neutralized and coagulation took

TABLE 1

"Before" indicates used oil as taken from ship.
"After" indicates the same oil after the special treatment.

Test No. . .	I		II		III		IV	
	Before	After	Before	After	Before	After	Before	After
Specific gravity at 60 deg. F..	0·9360	0·9074	0·923	0·909	0·924	0·905	0·942	0·908
Water, per cent	Trace	Nil	0·6	Trace	2·9	Nil	8·00	Trace
Carbon, etc., insoluble in benzol, per cent . .	2·00	Trace	2·80	Trace	2·5	Nil	4·87	Trace
Acid value equivalent to SO_3 per cent .	0·15	0·028	1·11	0·90	0·21	0·07	*	*
Ash, per cent .	0·65	0.07	0·73	Trace	*	*	1·90	Trace
Flash point (Pensky Martens) deg. F.	380	396	*	*	*	*	*	*
Viscosity,‡ sec. Redwood at 70 deg. F. .	4,356	1,743	†	1,700	2,048	1,526	2,231	1,440
Viscosity,‡ sec. Redwood at 100 deg. F. .	1,355	570	†	570	687	540	830	473
Viscosity,‡ sec. Redwood at 140 deg. F. .	414	187	†	184	219	184	315	154

* Undetermined.
† Undetermined owing to excess of solids.
‡ The viscosity of the unused oil was unknown.

place within a short period, clots being formed which were large enough to settle out in a few hours under the action of gravity or instantaneously in a centrifuge. The latter method was preferable, not only because continuous operation was possible, but also because a higher yield was secured together with greater purity.

It was a notable feature of the process that only 0·5 per cent of added water would deal satisfactorily with up to 4 per cent of carbon. The process was carried out by heating the oil and adding the necessary water as it passed through a special coagulating device, known as the "Colloid Coagulator", after which the mixture flowed through a tank fitted with baffle plates to prevent short-circuiting. The tank ensured a delay of about ten minutes to permit the formation of clots and the mixture then passed into the centrifugal separator.

If an output of about 15 gal. per hr. was required, an ordinary separator of the internal cone type would give satisfactory results. It was however, necessary to modify the tinware casing in order to enable the dirt to be carried away continuously. For higher outputs, it was essential to employ the "Ultra Centrifuge" which had been specially designed for the purpose. The continuous discharge of the dirt was accomplished by injecting a small stream of hot water into the separator.

Some typical results obtained by the process with different oils removed from various Diesel-engined ships fitted with ordinary centrifuges were given in Table 1.

Mr. H. Higinbotham (Messrs. E. G. Acheson, Ltd.) in a written reply, pointed out that the results quoted by Professor Norlin in regard to the use of oil containing colloidal graphite were contradictory to those obtained in Great Britain, America, and most other countries. The evidence showing that colloidal graphite of known purity and fineness reduced engine wear included extensive bench tests by independent authorities, and road tests totalling more than 100,000 miles, quite apart from the general use of the product by motorists and transport fleet operators. Furthermore, the tests referred to by Professor Norlin were open to criticism because, amongst other things:—

(1) The operating conditions admitted of variations in results.
(2) The mileage and measurements of cylinder wear were so small as to admit of experimental errors.
(3) There was no evidence that both cars were completely run-in, thus introducing a further margin of error.
(4) The iron content of the oil was not consistent with the wear figures.

Mr. Ricardo (p. 608) pointed out that the results of road tests were erratic and the writer therefore suggested that the test referred to by

Professor Norlin could not be considered as a serious contribution to the subject of engine wear.

Mr. D. R. Pilkington (p. 646) asked whether the presence of colloidal graphite on the upper part of the cylinder walls was likely to contribute to sludge. It could be stated in reply that there was no evidence to show that that was so. On the contrary, it had been shown that colloidal graphite discouraged the sticking of piston rings, particularly in Diesel engines. Considering the graphited surface in relation to sludge formation, that surface was chemically inert and its structure did not permit of a key for adherent particles. Tests had shown that colloidal graphite did not contribute to sludge formation in the sump oil. In fact, there were reasons why the opposite effect was to be expected and had in practice been recorded. Furthermore, it had been suggested that a metallic surface could act in certain circumstances as a catalyst in the formation of sludge in lubricants, so that the interposition of an inert graphite surface between a bearing face and the oil film might be desirable from that aspect alone. The writer had information showing that colloidal graphite, if anything, discouraged the formation of hard asphaltums in an oil.

The colloidal graphite to which the writer referred in his paper would pass through filters of the felt type. Black carbonaceous deposits in filters were often confused with colloidal graphite. Edge filtration was claimed to remove the finest particles of most materials, but some filters of that type could be set so that they did not remove colloidal graphite from the oil during the early life of the engine.

Colloidal graphite was accepted as being of value in reducing cylinder wear, particularly that due to frequent stopping and starting. Available information indicated that colloidal graphite enabled an engine to give its optimum power output in a shorter time than would be possible if metal pick-up or overheating was experienced. Reference had already been made by Professor G. I. Finch and Professor N. K. Adam to the ability of a graphited surface to prevent metal-to-metal contact that might ultimately lead to seizure.

Mr. Pilkington's remark about the suitability of graphite in the absence of an explosion was difficult to reconcile with the information concerning its value as an upper cylinder lubricant. Colloidal graphite could withstand higher temperatures than the heaviest oil or grease, and was, therefore, adapted to such work.

Colloidal graphite remained satisfactorily dispersed in lubricating oil. That it was extensively used during the running-in of automobile engines on the roads showed that the fineness of the graphite particles achieved its object in that respect. Herr Ostwald had referred to the suspension of colloidal graphite in fuel, and stated that it was quite

satisfactory with respect to the carburettor and fuel pump. In practice no difficulties were experienced with the addition of a lubricant containing colloidal graphite to the fuel. Its use in Great Britain was widespread and it was approved by certain engine manufacturers.

The views concerning graphite as a lubricant in locomotives put forward by Professor Nordmann and Herr Robrade had been dealt with by Mr. Brillié, and the writer was in agreement with Mr. Brillié's views in that connexion.

Mr. D. F. PILKINGTON (Transport Manager, Lancashire Associated Collieries) wrote to ask whether the presence of colloidal graphite on the upper part of the cylinder walls was likely to lead to the formation of sludge. It might reduce wear, but if it caused ring-sticking, it was of doubtful value. With colloidal graphite, felt filters were slightly choked, but not gauze filters, and it would be interesting to know what would happen when bypass filtration with edge filters was employed. Commercial users were waiting for a lead regarding the use of colloidal graphite. The experience of the German railways was rather extreme, as the general experience was that graphite was excellent for running-in bearings or surfaces where there was no explosion. In the paper by Herr Ostwald, it was stated that it was difficult to maintain colloidal graphite in dispersion in the fuel and it would be of value to know whether it remained in suspension in lubricating oil.

With regard to corrosion, it would seem better to add some "anti-corrosive" to even a poor oil than use a good oil without such an addition. The difficulty was that the ordinary driver was likely to be inaccurate in adding tablets to the petrol tank, so that it was better to treat the oil directly. That applied to any type of addition, for the human element could not be neglected.

They were told to use heavy oils in summer and thinner oils in winter, but when did summer begin in England? That meant that two grades had to be kept, with the possibility of contamination. To overcome that trouble, instead of using an oil of a viscosity of 12,000 sec. Redwood at 40 deg. F. in summer and an oil of 8,000 sec. Redwood viscosity in winter, by using a solvent-treated oil of a viscosity of 8,000 sec. at 40 deg. F., that oil was better at 212 deg. F. than the ordinary type with a viscosity of 12,000 sec. at 40 deg. The temperature of 40 deg. F. was given because most of the 150 petrol-engined lorries with which the writer was concerned were in cold garages. Now it was stated that solvent-treated oils caused more ring sticking than ordinary oil, but possibly that applied only to Diesel engines. There seemed, however, a great doubt in the minds of lubrication experts as to the advisability of using solvent-treated oils. One oil firm, which could supply either

ordinary or solvent-treated oils, had warned the writer against the latter as they had heard such conflicting reports of those oils from different parts of the world, and claimed that blended oils had "staying power", whereas a solvent-treated oil tended to "die out". Could any authoritative statement be made to disperse the doubts in the minds of commercial users?

He would like to know what the filter makers thought of the statement by Professor Trillat (see vol. 2) implying that by the use of filters oiliness was lost, though it was wanted.

Mr. W. A. STANIER, replying to Mr. H. N. Bassett (p. 638), stated that no difficulty had been experienced with the formation of deposits in the annular space round the periphery of the piston valve liners. The groove was $\frac{3}{4}$ inch deep and $1\frac{3}{8}$ inches wide, while the liner was provided with 8 equally spaced holes, $\frac{1}{4}$ inch in diameter. The immunity from trouble was due to the fact that that method of applying the oil was used only on engines fitted with atomized lubrication.

Mr. J. G. VINEALL (Ragosine Oil Company), in a written contribution, wrote that oxidation tests in general had received very severe criticism, but it had been overlooked that they had served a very useful purpose in raising the general level of the quality of engine lubricants during recent years with regard to their stability to heat and oxidation. The differences between performances in engines and results on test which had been stressed in various papers were not of great significance as the oils were, in all cases, good-quality oils of high stability. The real value of the Air Ministry test, and similar tests, consisted in distinguishing between oils which might be and oils which were not suitable for engine lubrication. The former class consisted of oils which gave comparatively low viscosity increases and coke increases, i.e. figures of the order of the Air Ministry test limits, and also yielded oxidized oils which were still fluid and free from deposits. The second class consisted of oils which gave very high oxidation figures and yielded oxidized oils which might be semi-solid and full of visible oxidation products of an asphaltic nature.

The advent of tests such as the Air Ministry test had made it possible to discriminate between oils which were fit for engine lubrication and those which yielded high carbon and sludge deposits. That had forced the producers of oil to evolve more complete methods of refining their lower-grade oils, with the result that, instead of being marked for purposes for which they were not suited, those asphaltic oils were subjected to processes such as solvent refining and more stable lubricants were produced which gave better results in the engine.

Eng. Lieut.-Commander A. CYRIL YEATES (Crossley-Premier Engines, Ltd.) wrote, with regard to the query raised by Mr. J. F. Alcock (p. 635), that the liner wear records were obtained from land engines operating under widely varying conditions. Many of them were in dusty localities, but in most cases those engines were well protected by air-conditioning plants or filters, whilst others were operating in very humid climates. Plotting the curve in the way suggested by Mr. Ricardo, but on a larger scale, the kink referred to by Mr. Alcock was not so apparent, although the linear relation between the frequency at starting and the rate of wear did not actually occur until the number of starts exceeded 50 per 1,000 hours of running time.

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